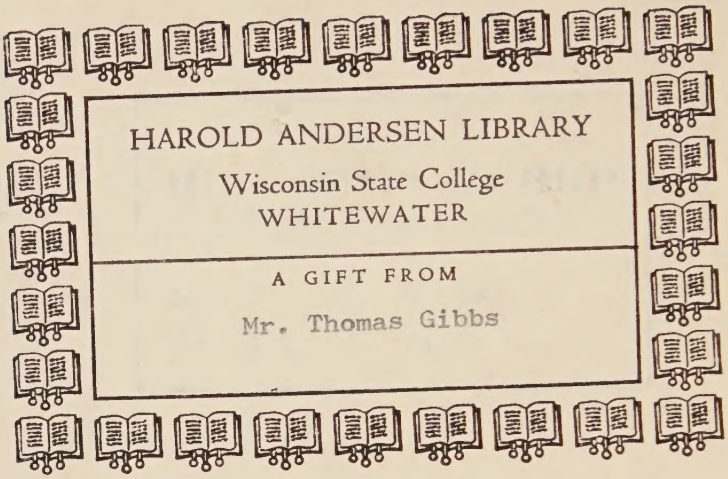
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STEAM, AIR, AND GAS POWER

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PREFACE

THIS book has been prepared as an elementary text for use in courses dealing with Heat Engineering, where only a limited amount of work in the subject can be included. The work is intended for students taking only a single course, for those registered in an accompanying Mechanical Engineering laboratory course, or for those who will register in an experimental laboratory course later in the curriculum.

The aim has been to describe briefly and clearly typical and representative equipment, and to explain the theory of such machines and devices. Due to the great number of products in the market, only a few illustrating the points under consideration could be included. The mathematical calculations involved are of the simplest order and illustrative problems have been included wherever necessary. The subject matter dealing with Steam Engines has been reduced to a minimum; much material formerly included in similar textbooks is omitted.

The authors express their thanks to Professor G. A. Goodenough for permission to include data from his work, "Properties of Steam and Ammonia," and for aid in preparing certain portions of the text. The authors also thank Professors A. C. Willard, A. P. Kratz, and J. A. Polson, and Dr. John A. Goff for assistance rendered in reading and commenting on the text material. Appreciation of the cooperation received from the various manufacturers, whose equipment is illustrated, is expressed and wherever possible the use of such illustrations has been acknowledged.

W. H. S.
H. E. D.

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STEAM, AIR, AND GAS POWER

CHAPTER I

INTRODUCTION

1. Heat.—A body possessing heat has its molecules in a state of motion, the movement increasing as the body is heated and decreasing as it is cooled. Thus heat may be considered a form of kinetic energy. The amount of heat contained in a body, above any reference point, depends upon the quantity (weight) of the substance, its unit capacity (specific heat), and its temperature.

2. Temperature.—Temperature is a measure of the intensity of heat as distinguished from the quantity of heat. Temperature may be expressed as *the thermal state of a body considered with reference to its ability to transmit heat to other bodies.*

Instruments for measuring relatively low temperatures are called *thermometers*, while those used for measuring high temperatures (above 1000 deg. fahr.) are designated as *pyrometers*. The mercurial thermometer is the only type of thermometer used in heat engineering work and consists of a glass tube of very small uniform bore hermetically sealed at the upper end, terminating at the lower end in an enlargement of the bore which is filled with mercury, see Fig. 1. The space within the tube not filled by the mercury is usually a vacuum in the case of thermometers for measuring ordinary temperatures. For higher temperatures (above 200 deg. fahr.) and for greater accuracy at ordinary temperatures the space is filled with an inert gas, nitrogen under pressure, to prevent vaporization of the mercury.

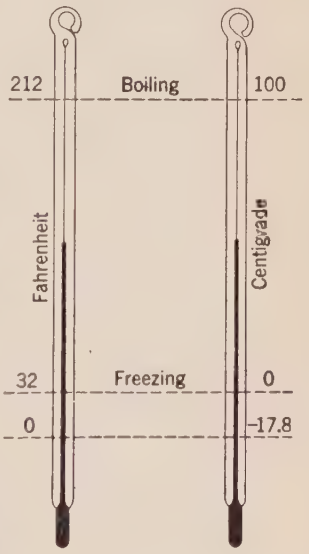

FIG. 1.—Thermometer scales.

The position of the top of the mercury column when the lower part of the thermometer is immersed in freezing water is called the *freezing point;* this is marked on the glass tube itself or on a scale attached to the tube. The position of the top of the mercury when the lower part of the thermometer is immersed in boiling water at "standard atmospheric pressure" (14.7 lb. per sq. in.) is called the *boiling point;* this is marked in the same way as the freezing point. The range between the freezing point and the boiling point is called the *fundamental interval* of the temperature scale.

On the centigrade scale the fundamental interval is divided into 100 equal parts, the freezing point is taken as zero and the boiling point as 100 deg. On the fahrenheit scale the fundamental interval is divided into 180 equal parts, the freezing point is taken as 32 deg. and the boiling point as 212 deg. The relation between the two temperature scales is shown diagrammatically in Fig. 1. If t_c denotes a temperature on the centigrade scale, and t_f denotes the same temperature on the fahrenheit scale, then

$$t_f = \tfrac{9}{5}t_c + 32 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and

$$t_c = \tfrac{5}{9}(t_f - 32) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

3. Absolute Temperature.—Experiments show that the amount by which air, hydrogen or any other so-called perfect gas expands per degree fahrenheit of increase in temperature—as in passing from 32 to 33 deg. fahr. at atmospheric pressure—is about $\frac{1}{491.6}$ of the original volume. When the scale of the gas thermometer is taken as the standard it may be shown, when 491.6 cu. in. of gas at 32 deg. fahr. are heated under constant pressure to 33 deg., that the volume will be 492.6 cu. in. Similarly, if the gas is cooled from 32 deg. fahr. to 31 deg. its volume changes from the original 491.6 to 490.6 cu. in. If the same law could be held to apply down to the lowest temperatures, the volume will become zero at 491.6 degrees below 32 deg. fahr. or at a temperature of − 459.6 deg. fahr. (practically − 460 deg.).

If the temperature is reckoned not from the freezing point but from a zero which is 491.6 fahrenheit degrees below freezing, the volume of a gas when heated under constant pressure is proportional to the temperature reckoned from that zero called the *absolute zero of temperature.* This absolute zero is sometimes defined as the point which marks the absence of heat energy or molecular motion in a body.

Denoting the temperature on the ordinary scale by t and the corresponding temperature reckoned from the absolute zero by T,

$$\text{On the fahrenheit scale, } T_f = t_f + 460 \quad . \quad . \quad . \quad . \quad (3)$$

$$\text{On the centigrade scale, } T_c = t_c + 273 \quad . \quad . \quad . \quad . \quad (4)$$

The "thermodynamic" scale of temperature starts from a zero which is absolute in the sense that no lower temperature can exist; the zero of the thermodynamic scale coincides with the absolute zero of the gas scale.

4. Measurement of Heat.—The quantity measurement of heat is made in terms of the *British thermal unit* (B.t.u.), which is defined as $\frac{1}{180}$ of the heat required to raise the temperature of one pound of water from 32 to 212 degrees fahrenheit. Thus the British thermal unit is the average amount of heat required to raise the temperature of a pound of water through one degree fahrenheit, between the conditions of freezing and boiling at atmospheric pressure (14.7 lb. per sq. in.).

5. Specific Heat.—The *specific heat* (sometimes called thermal capacity) of any substance is the amount of heat required to raise the temperature of a unit weight of the substance one degree. The specific heat of all substances varies with temperature, and since all substances vary in volume or pressure with temperature changes, it is necessary to distinguish between specific heats at constant volume and at constant pressure, expressed ordinarily as c_v and c_p respectively.

Because of the low coefficients of expansion for liquids and solids, there is but little difference in the specific heat at constant volume and that at constant pressure. With gases, on the other hand, the difference is appreciable. When heat is added to a gaseous substance its volume may be kept constant, in which case no external work is done; or the gas may be allowed to expand during the addition of heat, the pressure being kept constant. The specific heat at constant volume, therefore, is always less than that at constant pressure by an amount equal to the heat required to do the work of expansion against external pressure.

In both specific heats, at constant volume and at constant pressure, it is necessary to distinguish between *instantaneous* and *mean* specific heat. The instantaneous specific heat of a substance is the amount of heat that must be added to a unit weight of such material at some definite temperature to increase its temperature one degree under given conditions of volume and pressure. The mean specific heat of a substance, over a given temperature range, is the average quantity of heat required to raise the temperature one degree of a unit weight of the substance under the conditions of volume or pressure which exist. In

practically all boiler and combustion work, the mean specific heats are used.

If a pound of water is considered as a standard, the mean specific heat over a temperature range of 32 to 212 deg. fahr. from the definition of a B.t.u. is unity. The specific heats of air at constant volume (c_v) and at constant pressure (c_p) vary with the temperature. A number of determinations of these values have been made and generally accepted values for the mean specific heats are:

c_v for air, 32 to 400 deg. fahr.	0.1739
c_p for air, 32 to 400 deg. fahr.	0.2425
Ratio of the specific heats $\frac{c_p}{c_v}$	1.395

6. Energy, Work, and Power.—Energy is the capacity for doing work and the ability to overcome resistance. Energy exists in many forms as heat, mechanical, electrical, chemical, etc. Several units are used in measuring energy and each type of energy has its characteristic unit of measurement, for heat energy the unit is the B.t.u. and for mechanical energy the *foot-pound.* The latter is defined as the capacity for lifting one pound weight through a vertical distance of one foot.

Potential energy is energy due to the relative position of one body to another, or of one part of a body to another part, when the two bodies or parts of the same body are under the action of a force or forces tending to alter their relative positions. For example, a body which is allowed to fall toward the earth may be made to do work; hence, before it begins to fall it possesses potential energy, because if allowed to resume its unrestrained form the body can be made to do work. Likewise, compressed air possesses potential energy. The energy stored in a piece of coal is potential energy and under favorable conditions the atoms of the constituents of the coal and the atoms of the oxygen of the air will rush together and produce heat, which may be converted into work.

Kinetic energy is energy due to the motion of a body. Ten pounds of water at rest at a height of 100 feet above sea level possess 1000 ft.-lb. of potential energy. If this water is allowed to fall freely to sea level, without doing work on the way and without friction or other losses, it will in any position of its fall possess 1000 ft.-lb. of energy; as it descends the potential energy will decrease and the remainder of the 1000 ft.-lb. will be stored in the water as kinetic energy. When the water has fallen 25 feet its potential energy will be 750 ft.-lb. and its kinetic energy will then be 250 ft.-lb. Heat is a form of kinetic energy, see Art. 1.

When a force acting on a body causes that body to move, the force is said to do *work*. In general, work is the overcoming of resistance through a distance. In measuring mechanical work the unit which is used by engineers is the work done when a force of one pound acts through a distance of one foot, this unit is the *foot-pound*. Thus, if P represents the magnitude of the force and dx the distance through which it acts,

$$\text{Work} = Pdx. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

If P is in pounds and dx is in feet, the work will be expressed in foot-pounds. If P is in pounds per sq. ft. and dx in cu. ft., the work will also be in foot-pounds.

The working *power* of any agent depends on the amount of work which it can do in a given time. A heat engine or any working agent is said to be developing one *horsepower* when it can do 33,000 foot-pounds of work in one minute. Using the same notation as for equation 5 and representing by y the time in minutes required to do the work, then

$$hp = \frac{Pdx}{33{,}000y}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The unit of electrical power is the *watt*, which is the power of an electrical current of one ampere flowing under an electromotive force (electrical pressure) of one volt. Thus, volts times amperes equal watts. A horsepower is equal to 746 watts and one kilowatt equals 1000 watts or 1.34 horsepower.

7. Absolute Pressure.—Barometers are instruments used to measure the pressure of the atmosphere, such pressure being commonly called the barometric or atmospheric pressure. There are two principal types, the mercurial barometer and the aneroid barometer. The mercurial barometer is the type generally used and recommended for accuracy. Fig. 2 is a diagrammatic illustration of the Fortin type, as used by the U. S. Weather Bureau. The instrument consists of a glass tube, about $\frac{1}{4}$-in. internal diameter and about 3-ft. long, closed at one end and inverted over a small cistern. The cistern and part of the glass tube are filled with pure mercury. Above the mercury in the glass tube is a high vacuum, giving essentially zero pressure on top of the column. The atmospheric pressure depresses the mercury in the cistern, so that the column of mercury in the glass tube measured above the level in the cistern exactly balances the atmospheric pressure. The exact height x, of the top of the mercury column in Fig. 2 is read by means of a scale

fastened to a brass shield which surrounds the glass tube. A sliding vernier is provided so that precise readings may be obtained. A thermometer is always attached to give the temperature of the column.

The atmosphere exerts a variable pressure, depending upon the altitude and the weather conditions. *Standard atmospheric pressure* is defined as the pressure of a column of pure mercury, 29.92 in. in height, at a temperature of 32 deg. fahr. A cubic inch of mercury at 32 deg. fahr. weighs 0.491 lb. and for ordinary temperatures this weight may be used with sufficient accuracy, or a column of mercury 1 in. high exerts a pressure of 0.491 lb. per sq. in. Hence, a reading in inches of mercury is changed to pounds per square inch by multiplying the height of the mercury column by 0.491. The standard atmospheric pressure is 14.7 lb. per sq. in.

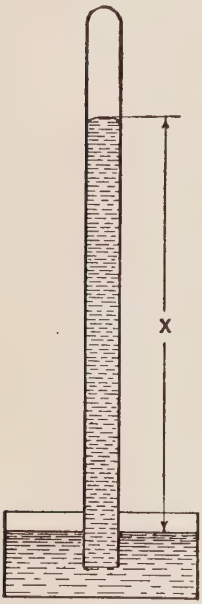

FIG. 2.—Mercurial barometer.

Gages indicating pressures above that of the atmosphere are called pressure gages (see Fig. 46), and those showing pressures less than atmospheric are known as vacuum gages. The gage pressure plus the atmospheric pressure, both expressed in the same units, is called the *absolute pressure*, i.e., absolute pressure is the pressure above zero pressure. For cases where a measured pressure is below atmospheric, the pressure indicated by the corrected reading of a *U*-tube (case A, Fig. 120) or a vacuum gage is called *vacuum*, usually measured in inches of mercury, and may be expressed with a minus sign. The absolute pressure in such cases is the difference between the barometric reading and the vacuum gage reading, or the algebraic sum of the two readings.

Example.—Find the absolute boiler pressure when the pressure gage shows 148 lb. per sq. in. and the barometer reads 29.45 in. of mercury.

Solution.—Atmospheric pressure = $29.45 \times 0.491 = 14.46$ lb. per sq. in. hence absolute boiler pressure = $148 + 14.46 = 162.46$ lb. per sq. in.

Example.—Find the absolute pressure in a condenser when the vacuum gage indicates 28.20 in. of mercury and the barometer reads 29.58 in. of mercury.

Solution.—Absolute condenser pressure = $29.58 - 28.20 = 1.38$ in. of mercury; hence absolute pressure = $1.38 \times 0.491 = 0.678$ lb. per sq. in.

PROBLEMS

1. Change the following temperatures on the centigrade scale into their corresponding temperatures on the fahrenheit scale, 1500, 550, 102, 22, 8, − 10.

2. Express the following temperatures on the fahrenheit scale in degrees centigrade, 1742, 572, 203, 77, 25, 0, − 4.

3. Find at what temperature the reading of a thermometer on the fahrenheit scale will be the same as on the centigrade scale.

4. Change to the corresponding absolute temperatures, 480 deg. fahr., 100 deg. cent., 32 deg. fahr., − 10 deg. cent., − 50 deg. fahr.

5. (*a*) Find the B.t.u. required to heat 5 lb. of air from 70 deg. fahr. to 350 deg. fahr., the pressure being constant. (*b*) How many B.t.u., if the volume had been constant?

6. How many pounds of air will absorb 237.5 B.t.u. at constant pressure, if the temperature increases from 40 deg. fahr. to 240 deg. fahr.?

7. How many B.t.u. must be removed to lower the temperature of 8 lb. of water from 190 deg. fahr. to 40 deg. fahr.? Assume specific heat unity.

8. A pump raises 11,000 lb. of water 30 ft. every minute. (*a*) Find the work done. (*b*) What is the horsepower developed?

9. If the pressure gage on a boiler reads 202 lb. per sq. in. and the barometer reads 29.28 in. of mercury, what is the absolute pressure in the boiler in pounds per square inch?

10. The vacuum gage on a condenser reads 26.4 in. of mercury and at the same time the barometer reads 29.67 in. of mercury. What is the absolute pressure in the condenser in pounds per square inch?

CHAPTER II

THERMODYNAMIC PRINCIPLES

8. First Law of Thermodynamics.[1]—The principle of the "Conservation of Energy" in relation to heat and mechanical work constitutes the First Law of Thermodynamics. When heat is produced from the expenditure of mechanical work a definite quantity of heat appears for every unit of work done; and conversely, when heat is transformed into mechanical work, the same quantity of heat is transformed for every unit of work produced.

When the brakes are applied to the wheels of a moving train, in order to bring it to rest, the kinetic energy of the train is converted into heat at the rubbing surfaces of the brakes and wheels; and if the wheels skid, heat is produced at the rubbing surfaces of the wheels and rails. This example illustrates the conversion of mechanical work into heat. The reverse transformation takes place in a heat engine when the heat produced by the combustion of fuel is converted into work by the engine.

Recent experiments [2] have shown that 778.6 (about 778) ft-lb. of work are spent in producing a unit of heat (B.t.u.); this quantity is called the *mechanical equivalent of heat* and is generally represented by the symbol J. The foot-pound is a unit derived from considerations of force and distance involved in the performance of mechanical work; the B.t.u. a unit derived from considerations of the change of temperature due to heating. That there is a constant relation between the two units (namely, one B.t.u. = 778 ft.-lb.) simply means that mechanical work can be measured in B.t.u. as well as in foot-pounds, or that a quantity of heat can be measured in foot-pounds as well as in B.t.u. The symbol A is used for the reciprocal of J, i.e., $A = \frac{1}{J}$.

9. Second Law of Thermodynamics.—The first law states a relation that must be satisfied in any transformation of energy. The second law gives information regarding the possibility of transformation and

[1] Thermodynamics is defined as that branch of science which deals with the transformations of energy (especially heat) into mechanical work.

[2] "Report on Progress in Steam Research at the U. S. Bureau of Standards," by Osborne, Stimson and Fiock. *Mechanical Engineering*, Feb., 1928, p. 152.

the availability of a given form of energy for conversion into mechanical work. The Second Law of Thermodynamics has been stated in many ways, but essentially it is the law of the degradation of energy. Clausius' form of the second law is: It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.

The second law states in effect, that heat will not pass automatically from a colder to a hotter body. Heat can be forced to pass up, as in the action of a refrigerating machine, but only by applying an "external agency" to drive the machine. A heat engine acts by letting heat pass down from a hotter to a colder body, not by direct conduction one to the other but by making a medium alternately take in heat from the hot body and reject heat to the cold body. Thus the medium undergoes expansions and contractions in which its pressure is, on the whole, greater during expansion than during contraction; and in this way a *part* of the heat that is passing down through the engine is converted into mechanical work.

10. Working Substance.—The working substance is the medium or vehicle by which heat passes through a heat engine. The substances commonly employed are air, water and water-vapor (steam). Because the working substance has a capacity for taking in heat, it can act as a vehicle for conveying heat from one temperature to another. In this process volume changes occur and by means of these changes on the part of the working substance, the heat engine does work. The working substance receives heat by being brought into contact with some source of heat at a comparatively high temperature; it transforms a small part of this heat into mechanical work; and rejects the remainder to what may be called a receiver of heat, the latter at a comparatively low temperature. The greater part of the heat may be said to simply pass through the engine from the source to the receiver, becoming "degraded" as regards temperature in the process.

The working substance of a heat engine usually returns periodically to the same state of temperature, pressure, volume and physical condition. Each time this occurs, the substance is said to have passed through a complete *cycle* of operations. When the cycle is complete the substance contains whatever internal energy it had at the beginning, for the state of the substance is exactly the same. Consequently, any mechanical work that it has done must have been done at the expense of the heat which has been taken in during the cycle. The total amount of energy lost must be equal to the amount received during the cycle, for the internal energy (Art. 11) is the same at the end as at the beginning. The application of the First Law of

Thermodynamics for a complete cycle indicates

Heat taken in = Heat rejected + Work done by the substance.

11. The Energy Equation.—There is no means by which the total energy that a working substance contains can be measured. The changes in energy to be studied are: (1) changes which occur when the substance is taking in or giving out heat, (2) doing work, or (3) having work done upon it. If a substance takes in heat without doing work, its internal energy increases by an amount equal to the heat taken in. If work is done without taking in heat, the work is accomplished at the expense of internal energy, and the latter is diminished by an amount equal to the work done. When heat is being absorbed and the substance is at the same time doing work,

Heat added to the substance

= Increase of internal energy + Work done by substance,

and for a very small step in the process,

$$dQ = dU + dW. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Here dQ is considered as positive when heat is absorbed by the substance, hence a negative dQ indicates heat removed from the substance. Likewise, dW is regarded as positive when work is done by the substance as in expansion. As regards the units involved, each term may be measured in B.t.u. or foot-pounds, whichever are more convenient. However, if either one is to be specified it may be done by using the symbols A and J. Thus,

$$JdQ = dU + dW$$

would indicate that each term is to be measured in foot-pounds, whereas

$$dQ = dU + AdW$$

would indicate that B.t.u. are to be used.

At the end of a complete cycle there is no change of the internal energy U, and consequently for the cycle as a whole,

$$Q_1 - Q_0 = W$$

where $Q_1 - Q_0$ is the net amount of heat received, namely, the difference between the heat taken in and the heat rejected; and W is the mechanical work done in the complete cycle.

12. Work Done.—In an engine of the usual cylinder and piston type the working substance does mechanical work by changes of volume. Let the pressure of the working substance be P pounds per unit area while the piston moves a distance dx, this distance being taken infinitely small so that the pressure may be considered sensibly constant during the displacement. If the area of the piston is a sq. ft., the total force on it is Pa, and the work done is $Padx$. But $adx = dV$, the change of volume, hence

$$dW = PdV. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Substituting this value for dW in equation 7,

$$dQ = dU + PdV. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

A diagram may be drawn in which the relation of the pressure of any working substance to its volume is shown by rectangular coordinates, as in Fig. 3. The area $MABCN$ is obtained by the summation of elements of the type PdV and is expressed by the integral, $\int_1^2 PdV$. But from equation 8 this gives the external work done by the substance; hence the external work done is represented by the area $MABCN$. If the substance undergoes a complete cycle of change by expanding along curve ABC and by being compressed back through D to A, work is being done during expansion from A to C, equal to $MABCN$; and work is spent during compression from C through D to A, equal to the area $MADCN$. The net amount of work done by the substance during the cycle is equal to the algebraic sum of those areas; thus equal to the area of the closed figure or diagram $ABCDA$. However, if the process were such as to trace the figure in the opposite direction, the substance being expanded along ADC and compressed from C to A through B, the enclosed area would be a measure of the work expended upon the substance.

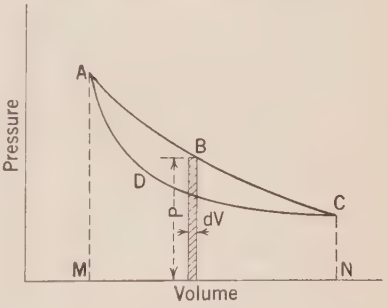

FIG. 3.—Pressure-volume diagram.

13. Gas Equations.—To illustrate the application of the preceding general statements regarding transformations of energy to specific cases, it is convenient to consider the so-called perfect gases because for these systems the relation between pertinent quantities (temperature, pressure and volume) is very simply expressed as in equation 14. Boyle's

law for a perfect gas is: *If the temperature is kept constant, the volume of a given weight of gas varies inversely as the pressure.* Thus,

$$Pv = C', \text{ a constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

or

$$P_1v_1 = P_2v_2 = P_3v_3 = - - -. \quad . \quad . \quad . \quad . \quad . \quad (11)$$

One statement of Charles' law is: *When a perfect gas receives heat at constant volume the pressure varies directly as the absolute temperature,*

$$\frac{P}{T} = C'', \text{ a constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

or

$$\frac{P_1}{T_1} = \frac{P_2}{T_2} = \frac{P_3}{T_3} = - - - . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

A perfect (or permanent) gas is a gas that is liquefied with difficulty and that obeys very closely the laws of Boyle and Charles. Such gases as air, oxygen, hydrogen, nitrogen, and helium are usually considered perfect gases.

Evidently the single relation that satisfies Boyle's and Charles' laws (compare equations 10 and 12) is

$$\frac{Pv}{T} = R, \text{ a constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

where v is the volume of 1 lb. of gas. Let M denote the weight of gas, then $V = Mv$, and substituting for v in (14),

$$PV = MRT \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

or

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} = - - -. \quad . \quad . \quad . \quad (16)$$

where P = absolute pressure in pounds per square foot.
V = volume of gas in cubic feet.
M = weight of gas in pounds.
R = gas constant (ft.-lb., per lb., per degree).
T = absolute temperature, deg. fahr.

The value of the constant R for a given gas may be determined, if the specific volume is known for any given condition of pressure and temperature. The specific volumes of various gases at atmospheric pressure (14.7 lb. per sq. in. abs.) and 32 deg. fahr. are: air, 12.39; oxygen, 11.21;

hydrogen, 177.9; and nitrogen, 12.80 cu. ft. per pound. Thus, for *air*,

$$R = \frac{14.7 \times 144 \times 12.39}{1 \times (459.6 + 32)} = 53.35$$

Example.—Find the volume of 15 lb. of air at 70 deg. fahr. under a pressure of 90 lb. per sq. in. gage.

Solution.— $PV = MRT$

$$V = \frac{15 \times 53.35 \times (460 + 70)}{144 \times (90 + 14.7)}$$

$$V = 28.13 \text{ cu. ft.}$$

Example.—If the air in the above problem expands to a volume of 142 cu. ft. and the final pressure is 5 lb. per sq. in. gage, what is the final temperature?

Solution.— $$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

$$\frac{144 \times (90 + 14.7) \times 28.13}{(460 + 70)} = \frac{144 \times (5 + 14.7) \times 142}{T_2}$$

$$T_2 = 503.4 \text{ deg. abs.}$$

and

$$t_2 = 503.4 - 460 = 43.4 \text{ deg. fahr.}$$

14. Joule's Law: Internal Energy.—In addition to the relation between temperature, pressure, and volume for perfect gases, additional information concerning the quantity U, the internal energy, is needed. That is, how the internal energy of a perfect gas varies with the temperature, pressure, and volume. Fortunately the desired relation is quite simple as the following experiment shows, because the internal energy of a gas does not vary with either pressure or volume, but only with the temperature. In 1844, Joule, an English physicist, experimented with the expansion of a perfect gas; he allowed the gas to expand in such a way that it received no heat, lost no heat, and did no external work. These precautions insured that the total internal energy could not change because $dQ = 0$, $dW = 0$, and hence, by equation 7, $dU = 0$.

The apparatus used by Joule is shown in Fig. 4. The vessel A was filled with air compressed to about 300 lb. per sq. in., and the air from B was exhausted as completely as possible. The two vessels were connected

by means of a pipe with a closed stop-cock C, and both vessels were immersed in a bath of water. The water in the bath was stirred and the temperature noted, then the stop-cock C was opened. After the gas had come to rest in the two vessels, the water was again stirred, and it was found that the temperature of the water in the bath had undergone no change.

It is natural to suppose that when the volume of a gas is increased its internal energy would also be increased because, with its molecules on the average farther apart, energy must have been supplied to separate them against their mutual attractions. If then the conditions of the experiment are such that no increase of internal energy is possible, the effect of separating the molecules (increasing the volume) must be counteracted, by a decrease of molecular motion, so that the mean kinetic energy of the molecules would be decreased. This retardation of the movement of the molecules would be manifested by a drop in temperature of the gas. The fact that no drop in temperature was observed is then to be interpreted to mean that the attractive forces between the molecules are negligible, that therefore no energy is required to separate them, and finally, that the internal energy of the gas as a whole is independent of the volume. Thus, a change of pressure and volume not associated with a change of internal energy does not alter the temperature. Or conversely, in any change of temperature *the change of internal energy* is independent of the relation of pressure to volume during the operation; it *depends only on the amount by which the temperature has changed*. Hence, in the energy equation $dQ = dU + dW$, the term dU (internal energy) is proportional to the change in temperature; whereas dW (external work done) depends upon the changes of pressure and volume, as explained in Art. 12.

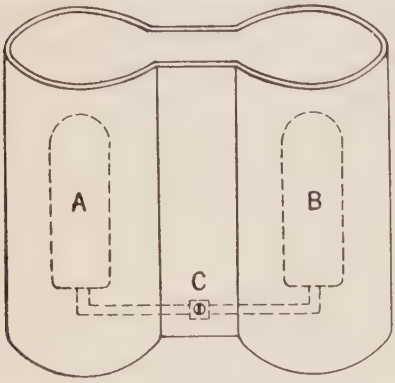

FIG. 4.—Joule's apparatus.

15. Specific Heats of a Gas.—When a gas is heated through a small change of temperature dT, the heat taken in (per pound) may be expressed in the form $c\,dT$, this expression being essentially a definition of the specific heat c for the particular manner in which the heating is accomplished (see Art. 5). When heating a unit quantity of a perfect gas at *constant volume* from temperature T_1 to T_2, the heat taken in is $c_v\,(T_2 - T_1)$ by definition of c_v. No external work is done, for the vol-

ume is constant and consequently all this heat goes to increase the internal energy; therefore for M pounds of a gas

$$dQ = dU = Mc_v dT \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

or (for constant volume)

$$Q = U_2 - U_1 = Mc_v(T_2 - T_1) \quad . \quad . \quad . \quad (18)$$

Heating M pounds of a perfect gas at *constant pressure*, the heat taken in is, by definition of c_p,

$$dQ = Mc_p dT \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

or (for constant pressure)

$$Q = Mc_p(T_2 - T_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

During this process external work is done, the amount being obtained from equation 8, the pressure being considered constant while the gas expands and

$$W = P(V_2 - V_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where V_1 and V_2 represent the volumes at the beginning and end of the operation respectively, and P is the pressure. But $PV_2 = MRT_2$ and $PV_1 = MRT_1$ and equation 21 becomes

$$W = MR(T_2 - T_1) \text{ in ft.-lb.} \quad . \quad . \quad . \quad . \quad . \quad (22)$$

or (for constant pressure)

$$W = AMR(T_2 - T_1) \text{ in B.t.u.} \quad . \quad . \quad . \quad . \quad (23)$$

Substituting in equation 7 and transposing,

$$U_2 - U_1 = Mc_p(T_2 - T_1) - AMR(T_2 - T_1)$$

or (for constant pressure)

$$U_2 - U_1 = (Mc_p - AMR)(T_2 - T_1) \quad . \quad . \quad . \quad . \quad (24)$$

Relation between c_p and c_v.—Equations 18 and 24 can be used to show an important relation between c_p and c_v for a gas. It is necessary to note that if T_1 and T_2 are the same in both cases, the changes of energy U will likewise be equal because Joule's law states that the temperature of a gas is all that need be known in considerations of its internal energy. The change in U from equation 18 equals change in U from equation 24 and

$$Mc_v(T_2 - T_1) = (Mc_p - AMR)(T_2 - T_1)$$

Simplifying,

$$c_v = c_p - AR \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

and

$$c_p - c_v = AR \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

The ratio of specific heats $\frac{c_p}{c_v}$ may be denoted by the symbol k, hence

$$c_p = kc_v \quad \text{and} \quad c_v = \frac{c_p}{k} \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Introducing the symbol k in equation 26,

$$c_p = \frac{kAR}{k-1} \quad \text{and} \quad c_v = \frac{AR}{k-1} \quad . \quad . \quad . \quad . \quad . \quad (28)$$

The specific heats of most gases change somewhat with the temperature. However, if the temperature range is not too large, it is permissible to use mean values. Table I gives mean values [3] of c_p, c_v, and k for several gases for a temperature range of 32 to 400 deg. fahr.

TABLE I

Mean Specific Heats of Gases

Gas	c_p	c_v	$k = \frac{c_p}{c_v}$
Air	0.2425	0.1739	1.395
Oxygen	0.2183	0.1563	1.397
Hydrogen	3.2000	2.2150	1.444
Nitrogen	0.2493	0.1785	1.397

As illustrations of the results thus far obtained in the special cases when either the pressure or the volume is held constant, consider the following cases.

Example.—Eight cubic feet of air are cooled at a constant pressure of 80 lb. per sq. in. abs. from 180 deg. fahr. to 100 deg. fahr. Find the external work done in foot-pounds, the change of internal energy in foot-pounds, and the heat abstracted in B.t.u.

[3] Calculated from equations given in Bulletin 139, Engineering Experiment Station, University of Illinois, 1924, "An Investigation of the Maximum Temperatures and Pressures Attainable in the Combustion of Gaseous Fuels," by G. A. Goodenough and G. T. Felbeck.

Solution.— $P_1V_1 = MRT_1$

$$80 \times 144 \times 8 = M \times 53.35 \times 640$$

$$M = 2.7 \text{ lb. of air}$$

From (22)

$$W = MR\,(T_2 - T_1)$$

$$W = 2.7 \times 53.35\,(560 - 640)$$

$$W = -11{,}524 \text{ ft.-lb.}$$

NOTE.—Negative sign of W indicates compression.

From (24)

$$U = (Mc_p - AMR)(T_2 - T_1)$$

$$U = \left[(2.7 \times 0.2425) - \frac{2.7 \times 53.35}{778}\right](560 - 640)$$

$$U = -37.57 \text{ B.t.u.} = -29{,}230 \text{ ft.-lb.}$$

NOTE.—Negative sign of U indicates loss of internal energy.

From (20)

$$Q = Mc_p\,(T_2 - T_1)$$

$$Q = 2.7 \times 0.2425\,(560 - 640)$$

$$Q = -52.4 \text{ B.t.u.} = -40{,}767 \text{ ft.-lb.}$$

NOTE.—Negative sign of Q indicates removal of heat.

Example.—A tank having a volume of 25 cu. ft. contains air at 14.5 lb. per sq. in. abs. and a temperature of 70 deg. fahr. How many B.t.u. will be required to heat the air to 150 deg. fahr.?

Solution.— $P_1V_1 = MRT_1$

$$14.5 \times 144 \times 25 = M \times 53.35 \times 530$$

$$M = 1.85 \text{ lb. of air}$$

From (18)

$$Q = Mc_v(T_2 - T_1)$$

$$Q = 1.85 \times 0.1739(610 - 530)$$

$$Q = 25.74 \text{ B.t.u.}$$

16. Adiabatic Processes.—Another special case to which the general gas equations apply, and which is of much importance practically, is that of an expansion or compression involving no transfer of heat to or from

the gas. Such a change of state is called *adiabatic*. From this definition it follows that the work which a gas does while it is expanding adiabatically is all done at the expense of its internal energy; and the work which is spent upon a gas to compress it adiabatically goes to increase its stock of internal energy. In actual heat engines the action is never strictly adiabatic, for there are always some exchanges of heat between the working substance and the surfaces of the cylinder and piston. Rapid expansion or compression approaches being adiabatic by giving little time for heat transfer.

In the adiabatic expansion of a gas, work is done; and since no heat is taken in or given out, there must be a decrease of internal energy equivalent to the amount of work done. Taking the general gas equation, $dQ = dU + PdV$, which applies to any small change of state, $dQ = 0$ when the change is adiabatic; and hence for adiabatic expansion,

$$PdV = -dU \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

where PdV is the work done and dU is the change of internal energy.

From equation 17, $dU = Mc_v dT$; hence from (29),

$$APdV = -Mc_v dT \text{ in B.t.u.}$$

But

$$P = \frac{MRT}{V} \quad \text{and} \quad \frac{AMRTdV}{V} + Mc_v dT = 0$$

or dividing by MT,

$$\frac{ARdV}{V} + \frac{c_v dT}{T} = 0$$

which gives by integration,

$$AR \log_e V + c_v \log_e T = \text{constant} \quad . \quad . \quad . \quad . \quad (30)$$

Writing $c_p - c_v = AR$ from (26) and dividing by c_v, which is constant,

$$\left(\frac{c_p}{c_v} - 1\right) \log_e V + \log_e T = \text{constant}$$

But

$$\frac{c_p}{c_v} = k,$$

thus,

$$k \log_e V - \log_e V + \log_e T = \text{constant} \quad . \quad . \quad . \quad (31)$$

Since $\frac{PV}{T} = MR = \text{constant}$, changing to log. form.

$$\log_e P + \log_e V - \log_e T = \text{constant} \quad . \quad . \quad . \quad . \quad (32)$$

Adding equations 31 and 32,

$$\log_e P + k \log_e V = \text{constant} \quad . \quad . \quad . \quad (33)$$

which gives

$$PV^k = \text{constant} \quad . \quad . \quad . \quad (34)$$

or

$$P_1 V_1^k = P_2 V_2^k = P_3 V_3^k = - - - \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

Combining equations 16 and 35,

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}} \quad . \quad . \quad . \quad . \quad (36)$$

In *any kind of expansion* of a gas the work done in expanding from volume 1 to volume 2 (from equation 8) is

$$W = \int_1^2 P dV$$

If the nature of the expansion is such that PV^n is constant, n being any exponent, then P at *any* point is $\frac{P_1 V_1^n}{V^n}$, P_1 and V_1 being the pressure and volume in the initial state. Then for expansion from V_1 to V_2,

$$W = P_1 V_1^n \int_1^2 \frac{dV}{V^n}$$

which gives by integration,

$$W = \frac{P_1 V_1^n (V_2^{1-n} - V_1^{1-n})}{1 - n}$$

or

$$W = \frac{P_1 V_1 - P_2 V_2}{n - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

Applying equation 37 to a gas expanding adiabatically, for which the exponent n assumes the value k (by equation 34),

$$W = \frac{P_1 V_1 - P_2 V_2}{k - 1} = \frac{MR(T_1 - T_2)}{k - 1} \quad . \quad . \quad . \quad (38)$$

NOTE.—Equation 38 expresses also the decrease of internal energy produced by adiabatic expansion.

Example.—Three cubic feet of air enter an air compressor on the suction stroke at atmospheric pressure (14.5 lb. per sq. in. abs.) and are compressed adiabatically to a pressure of 87 lb. per sq. in. abs. (*a*) Find the work done during compression. (*b*) What is the final temperature, if the initial temperature is 68 deg. fahr.?

Solution.—(*a*) From (35)

$$V_2 = 3\left(\frac{14.5}{87}\right)^{\frac{1}{1.4}} = 0.834 \text{ cu. ft.}$$

From (38)

$$W = \frac{144[(14.5 \times 3) - (87 \times 0.834)]}{0.40}$$

$$W = -10{,}460 \text{ ft.-lb.}$$

(*b*) From (36)

$$T_2 = 528\left(\frac{87}{14.5}\right)^{\frac{0.4}{1.4}} = 880.9 \text{ deg. abs.}$$

$$t_2 = 420.9 \text{ deg. fahr.}$$

17. Isothermal Processes.—Still another special case, to which the general gas equations apply and which is of practical importance, is that of an expansion or compression during which the temperature of the gas remains constant. Such a change of state is called *isothermal.* When a gas is expanding isothermally it takes in heat to maintain its temperature constant; it, therefore, must be in contact with an external source of heat. When it is being compressed isothermally the gas gives out heat, and must be in contact with a receiver which can take heat from the gas; the water jacket of an air compressor serves as a partial receiver of heat. If T is made constant in the equation $PV = MRT$,

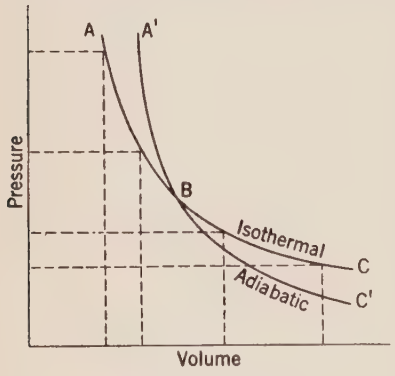

FIG. 5.—PV curves.

$$PV = \text{constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

or

$$P_1V_1 = P_2V_2 = P_3V_3 = - - - \quad . \quad . \quad . \quad . \quad (40)$$

This is the equation of the isothermal curve in PV coordinates and the curve is an equilateral hyperbola. Fig. 5 shows the isothermal (ABC) and adiabatic ($A'BC'$) curves drawn on the PV plane.

To find the external work done in the isothermal expansion of a gas from V_1 to V_2,

$$W = \int_1^2 PdV$$

and from (39)

$$P = \frac{P_1 V_1}{V}$$

from which

$$W = P_1 V_1 \int_1^2 \frac{dV}{V}$$

Integrating

$$W = P_1 V_1 (\log_e V_2 - \log_e V_1)$$

or

$$W = P_1 V_1 \log_e \frac{V_2}{V_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

But since

$$PV = P_1 V_1 \quad \text{and} \quad PV = MRT$$

then

$$W = MRT \log_e \frac{V_2}{V_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

where T is the absolute temperature at which the process takes place. This expression gives either the work that is done by a gas in isothermal expansion, or the work that is spent on the gas in isothermal compression.

During the isothermal expansion or compression of a perfect gas there is no change of internal energy, since there is no change of temperature (Art. 14). Hence, during isothermal expansion a gas must take in an amount of heat equivalent to the work done, from equation 9, $dQ = APdV$, and from equation 42, $dQ = AMRT \log_e \frac{V_2}{V_1}$; and during isothermal compression from V_2 to V_1 it must reject that amount of heat.

Example.—Three cubic feet of air under a pressure of 80 lb. per sq. in. abs. are expanded isothermally to a pressure of 20 lb. per sq. in. abs. (*a*) Find the work done during this expansion. (*b*) Find the heat added in B.t.u.

Solution.—(*a*) From (40)

$$V_2 = \frac{3 \times 80}{20} = 12 \text{ cu. ft}$$

From (41)

$$W = 144 \times 80 \times 3 \log_e \tfrac{1}{3}^2$$

$$W = 47{,}910 \text{ ft.-lb.}$$

$$(b) \text{ Heat added} = \frac{47{,}910}{778} = 61.6 \text{ B.t.u.}$$

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18. Reversible and Irreversible Processes.—When considering thermodynamic processes in which a working substance may be expanded or compressed, and may take in or give out heat, the conception of reversibility plays an important part. There are no reversible processes in nature since no process is entirely free from friction or heat conductions; but the limiting situation, when the factors contributing to irreversibility are indefinitely diminished in their effects, may be analyzed as if it actually existed as a reversible process. An expansion or compression, therefore, is *reversible* if the entire process or operation can be reversed. The result is that the working substance will assume in inverse order all the stages through which it had passed during the expansion or compression, and be in the same condition in all respects at each corresponding stage in both processes. A process which does not satisfy these requirements in any particular is an *irreversible* process.

Reversible expansion implies that there shall be no losses of mechanical effect from any kind of friction. Throttling is excluded such as occurs when a substance expands through a valve or other constricted opening into a region of lower pressure. When the kinetic energy of the flow is dissipated in whirls and eddies, the motion cannot be reversed. The expansion which occurred in Joule's experiment (Art. 14) is an example of an irreversible process. Any thermal contact between bodies at different temperatures involves an irreversible transfer of heat. The heat will not of itself flow back from the colder to the hotter body (Art. 9). A transfer of heat to or from any substance is reversible only if the substance is at the same temperature as the body from which it is taking heat or to which it is giving heat.

Neither the expansions and compressions nor the transfer of heat that occur in a real engine are ever strictly reversible; in fact, most of them are far from being reversible. But the study of an ideal engine, in which all the operations are reversible, is of fundamental importance in the science of thermodynamics as it furnishes a basis for the more critical analysis of actions in a real engine.

19. The Carnot Cycle.—Carnot, a French engineer, pointed out (in 1824) that the greatest possible amount of work was to be obtained by letting the heat pass from the source to a receiver, through an engine working in a reversible manner; reversible not only as regards its own internal actions but also as regards the transfer of heat to the engine from the source, and from the engine to the receiver. The engine described by Carnot is an engine each of whose operations is reversible, as explained in Art. 18.

The Carnot cycle, Fig. 6, consists of four successive operations: (1) Isothermal expansion, heat taken in at the temperature of the hot

source T; (2) adiabatic expansion, during which the temperature of the working substance falls from T to T_0; (3) isothermal compression, heat rejected at the temperature of the receiver T_0; (4) adiabatic compression, during which the temperature of the working substance is raised from T_0 to T. This completes the cycle by bringing the substance back to the condition in which it was assumed to be at the beginning of the first operation.

Consider an ideal engine in which a working substance may go through the operations of Carnot's cycle, Fig. 6. Imagine a cylinder and piston composed of non-conducting material, except the end of the cylinder which is a conductor. A hot body S serves as an infinite source of heat, kept at temperature T; N is a non-conducting cover; and R is a cold body of infinite capacity for receiving heat, kept at temperature T_0 which is lower than T. Suppose that S, N, or R can be applied to the end of the cylinder, as desired.

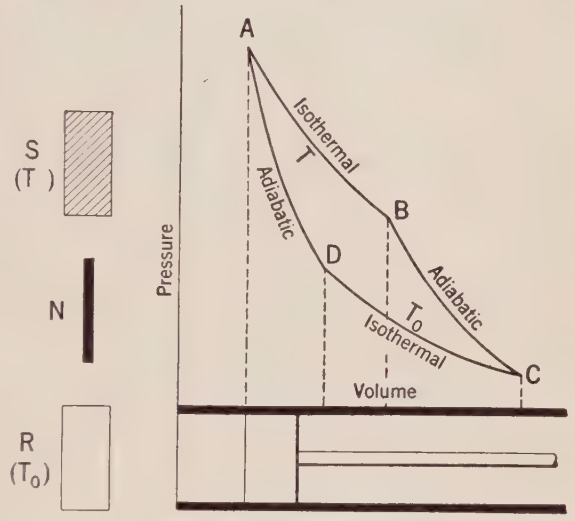


FIG. 6.—The Carnot cycle.

Throughout this chapter the properties of perfect gases have been used to illustrate some of the general thermodynamic principles, because of the ease with which these properties may be described analytically. Suppose, therefore, that the working substance used in the ideal Carnot engine is a perfect gas. Let the cylinder contain M lb. of a perfect gas at temperature T, volume V_a, and pressure P_a. Apply the hot source S to the conductor end of the cylinder and allow the frictionless piston to advance slowly to V_b and pressure P_b. The gas expands isothermally at T taking in heat and doing work. The non-conducting cover N is next placed against the cylinder end and the gas is adiabatically expanded to V_c and P_c; the piston advances and does work at the expense of the internal energy of the gas, while the temperature falls to T_0. Then the cold body or receiver R is applied to the cylinder end, the piston

is forced back slowly and the gas is compressed isothermally at T_0, since the smallest increase of temperature above T_0 causes heat to pass into R. Work is spent upon the gas and heat is rejected to the receiver R. Let this rejection of heat be continued until point D is reached, such that the next operation will complete the cycle. Remove R and apply N, continue the compression which is now adiabatic. The pressure and temperature rise and if point D has been properly chosen, when the pressure is restored to its original value P_a, the temperature will also have risen to its original value T. This completes the cycle.

During the adiabatic expansion from B to C, the relation between volume and temperature, from (36) is

$$\left(\frac{V_c}{V_b}\right)^{k-1} = \frac{T_b}{T_c} = \frac{T}{T_0}$$

and during the adiabatic compression from D to A, the relation is

$$\left(\frac{V_d}{V_a}\right)^{k-1} = \frac{T_a}{T_d} = \frac{T}{T_0}$$

hence

$$\frac{V_c}{V_b} = \frac{V_d}{V_a} \quad \text{or} \quad \frac{V_b}{V_a} = \frac{V_c}{V_d} \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

The transfers of heat in the four successive stages of the Carnot cycle are:

(1) Heat taken in from S, $Q = MRT \log_e \frac{V_b}{V_a}$. (See Art. 17).

(2) No heat taken in or rejected.

(3) Heat rejected to R, $Q_0 = MRT_0 \log_e \frac{V_c}{V_d}$. (See Art. 17.)

(4) No heat taken in or rejected.

The area $ABCD$, Fig. 6, enclosed by the four curves of the cycle, represents the mechanical work gained. The net amount of external work done by the gas is the difference between the heat taken in and the heat rejected in a complete cycle. Then from the first law of thermodynamics, $Q - Q_0 = W$. The *efficiency of the cycle* is the ratio of the work gained to the heat supplied from the source. Denoting the Carnot cycle efficiency by e_c,

$$e_c = \frac{Q - Q_0}{Q} = \frac{W}{Q} = \frac{MRT \log_e \frac{V_b}{V_a} - MRT_0 \log_e \frac{V_c}{V_d}}{MRT \log_e \frac{V_b}{V_a}}$$

But from (43), $\frac{V_b}{V_a} = \frac{V_c}{V_d}$; then

$$e_c = \frac{T - T_0}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

Suppose that there are two heat engines A and B, Fig. 7, one of which (A) is reversible, working between the same hot source of heat and cold receiver, and each producing the same amount of mechanical work. Let Q be the quantity of heat which A takes in from S. Now if A is reversed, this engine will, by the expenditure on it of the same amount of work, give to the hot source the amount of heat A formerly taken from S, namely Q. For this purpose set the engine B to drive A reversed; the work which B produces is just sufficient to drive A, and the two machines (B driving A) form together a self-acting machine unaided by any external agency. B takes heat from the hot body and A, which is reversible, gives back to S the amount of heat Q. Now if B could do its work by taking less heat than Q from S, the hot body would gain heat. But no work is being done on the system from the outside, nor is any heat supplied from other sources, so whatever heat the hot body gains must come from the cold body. Therefore, if B could do as much work as the reversible engine A, with a smaller supply of heat, it would be possible to arrange a self-acting machine through which heat would pass from a cold body to a hot body, thus violating the second law of thermodynamics.

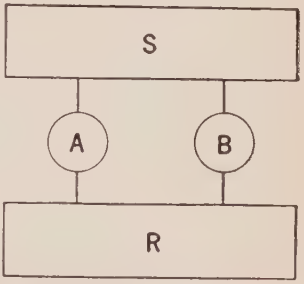


FIG. 7.—Two heat engines with source and receiver.

Thus B cannot do the same amount of work with a smaller supply of heat than a reversible engine; or, *no other engine can be more efficient than a reversible engine, when they both work between the same two temperatures of source and receiver.* If both engines A and B were considered reversible, the same argument shows that they are equally efficient. Hence, all reversible engines working between the same temperature limits are equally efficient, which necessarily implies that *the efficiency is independent of the working substance.*

Equation 44, $e_c = \frac{T - T_0}{T}$, measures the efficiency of *any* reversible heat engine; and therefore also expresses the largest fraction of the heat supplied that can possibly be converted into work by any engine when operating between temperature limits, T and T_0. If there is a supply of

heat at a temperature T and a means of disposing of heat at a minimum temperature of T_0, then there is no possibility of converting more than a fraction of the heat into work. This is the measure of the *best efficiency;* it is the theoretical limit beyond which the efficiency of a heat engine cannot go. No engine can surpass this standard; and the efficiency of a real engine is always less, as no real engine is strictly reversible.

20. Entropy.—The *availability* of a given quantity of heat energy for transformation into work is given by the efficiency of the ideal Carnot engine. Thus, if T is the temperature of the source of heat and T_0 the temperature of the coldest receiver available, then the efficiency is, $e_c = \frac{T - T_0}{T} = 1 - \frac{T_0}{T}$. If the heat dQ has been removed from the source, then $\left[1 - \left(\frac{T_0}{T}\right)\right] dQ$ has been transformed into work under ideal conditions, but the part $\left(\frac{T_0}{T}\right) dQ$ would be rejected. Now consider any thermodynamic system undergoing any desired change of state; divide its internal energy at any stage of the process into two parts which may be called the available part and the unavailable part, respectively. If, during the process considered, an amount of heat dQ is transferred to the system, the part $T_0\left(\frac{dQ}{T}\right)$ of this heat is unavailable and may be supposed to increase the unavailable energy of the system by this same amount. In general, unless the volume of the system is held fixed, mechanical work is done at the same time that heat is being added. But mechanical work has been used as available energy in all the considerations of the Carnot cycle and hence will not affect the *unavailable* part of the internal energy.

Dividing the increase of unavailable energy by T_0, a definition of what is called the entropy of the system is obtained. Thus,

$$dS = \frac{dQ}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

This equation is applicable for reversible processes only, as every process in the Carnot cycle is reversible; and the efficiency of the ideal Carnot engine was used as the underlying principle in the preceding paragraph. In a frictionless adiabatic process no heat is added or rejected and $\frac{dQ}{T}$ becomes zero from equation 45, hence the entropy does not change. The direction of a process, physical or chemical, that occurs of itself is such as will bring about an increase of unavailable energy of the

system. This principle is the foundation of the application of thermodynamics to chemistry. Electrical energy is rendered unavailable when transformed into heat in the transmission system. In the transformation of work into electrical energy or of electrical energy into mechanical work, some of the high-grade energy is transformed into heat and is rendered unavailable.

Experience shows that every actual physical process is irreversible and accompanied by losses. As a result the actual or irreversible process is accompanied by a decrease of the quantity of energy available for transformation into work, or an increase in unavailable energy. For an irreversible process the change of entropy is

$$dS = \frac{dQ}{T} + \frac{dQ'}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

in which dQ denotes the heat added by the substance from outside and dQ' the heat generated within the substance because of friction.

When a thermodynamic system undergoes any change, the sum of the entropy changes of the bodies which take part is the change in entropy of the system. The entropy of a system is the same whenever the system is in the same physical state. When a substance has passed through any complete cycle of operations its entropy at the end is identical with the entropy at the beginning, for the original state has been restored in all respects. This is true of an irreversible cycle as well as for a reversible cycle. Entropy, like energy, is relative and any expression for the entropy of a system contains an unknown constant of integration. Only differences of entropy are considered, therefore the constant of integration disappears by subtraction. The entropy changes of a substance are usually calculated per unit of mass, and numerical values of it are given in tables of the properties of steam and of other substances used in heat engineering. In assigning these numerical values it is customary to assume the entropy as zero at some standard state. Thus in the tables of properties of steam, the entropy of the liquid at 32 deg. fahr. is taken as zero.

21. Temperature-Entropy Diagrams.—In making use of entropy, it may be considered as a quantity having the property that the heat absorbed by a thermodynamic system is given by the expression

$$dQ = TdS$$

just as the mechanical work performed is

$$dW = PdV$$

The discussion (Art. 20) of the unavailable energy of a system was given, however, in order to establish entropy as a property of the system which returns to its original value when the system returns to its original state.

The area of the indicator diagram, from Art. 12, shown in Fig. 8, represents the external work done to some scale. Of equal interest is the graphical representation of the changes of entropy on a plane having the absolute temperature T as the ordinate and the entropy changes S as the abscissa; for a reversible process, the area enclosed represents the heat used throughout the complete cycle. The *temperature-entropy diagram* $A'B'C'D'$, in Fig. 9, represents the heat used throughout the

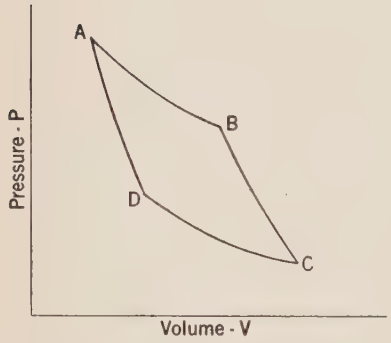


Fig. 8.—Indicator diagram.

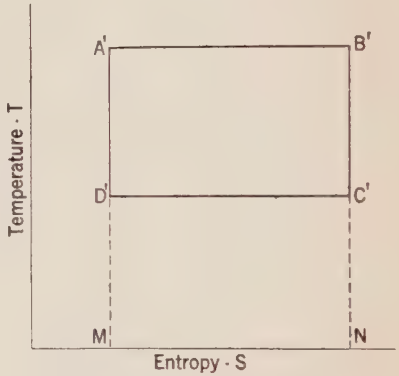


Fig. 9.—Temperature-entropy diagram.

complete cycle for which the indicator (PV) diagram, in Fig. 8, was drawn. During the isothermal expansion AB on the PV diagram, the change of entropy is represented by $A'B'$ on the TS diagram; for the adiabatic expansion BC, the entropy is constant and is shown by line $B'C'$. Similarly, during isothermal compression CD the change in entropy is $C'D'$, while the adiabatic compression DA is shown by line $D'A'$ on the TS diagram. The area $MA'B'N$ measured to the base line, which is the absolute zero of temperature, is the heat taken in from the source. The area $MD'C'N$ is the heat rejected to the receiver.

Let the curve AB, Fig. 10, represent a reversible change of state on the TS plane; the points A and B denoting respectively the initial and final conditions. The area $MABN$ or $\int TdS$ between the limits A and B represents the heat taken in while the substance changes in a reversible manner from the state represented by A to the state repre-

sented by B. Similarly, in changing reversibly from state B to state A, along line BA, the substance rejects an amount of heat measured by the area $NBAM$.

If the curve AB in Fig. 10 shows an irreversible change of state, the area $MABN$ represents the *sum* of the heat absorbed by the substance from outside and the heat generated within the substance because of friction. Should the process be frictionless, the area between AB and the S axis represents the heat absorbed by the substance. If, on the other hand, the process is adiabatic but not frictionless, then no heat is absorbed and the area represents the heat generated within the substance.

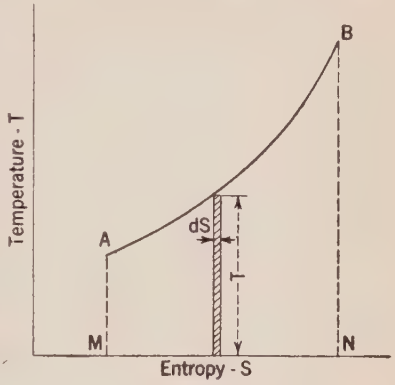


FIG. 10.—TS curve.

PROBLEMS

1. Determine the values of the gas constant R for oxygen, hydrogen, and nitrogen.

2. Find the volume of 5 lb. of air when at a pressure of 80 lb. per sq. in. abs. and a temperature of 170 deg. fahr.

3. If the air in problem 2 expands to a volume of 62 cu. ft. and the final pressure is 16 lb. per sq. in. abs., what is the final temperature in deg. fahr.?

4. A tank has a volume of 500 cu. ft. (*a*) Find the weight of air contained in this tank if the temperature is 70 deg. fahr. and the absolute pressure is 90 lb. per sq. in. (*b*) What will be the absolute pressure in lb. per sq. in., if the temperature is increased to 180 deg. fahr.?

5. A tank contains 2 lb. of air at a pressure of 100 lb. per sq. in. abs. and a temperature of 60 deg. fahr. The pressure in the tank should not exceed 300 lb. per sq. in. abs. and the temperature may rise to 100 deg. fahr. Find the weight of air that can safely be added to the contents of the tank.

6. The volume of a tank is 35 cu. ft. and contains air under a pressure of 90 lb. per sq. in. abs. at a temperature of 70 deg. fahr. Some of the air escapes until the pressure is 63 lb. per sq. in. abs. and the temperature is 63 deg. fahr. Find the volume occupied by the air removed from the tank at atmospheric pressure (29.53 in. Hg.) and 70 deg. fahr.

7. How many B.t.u. will be required to double the volume of 2 lb. of air at constant pressure from an initial temperature of 40 deg. fahr.?

8. If one pound of air at 40 deg. fahr. has its volume doubled at constant atmospheric pressure (barometer 29.22 in. Hg), (*a*) what is its final temperature; (*b*) how much external work is done during the expansion; and (*c*) how much heat must be supplied during the expansion?

9. A tank contains 50 cu. ft. of air at 100 deg. fahr. under a pressure of 110 lb.

per sq. in. abs. If the air in the tank receives 200 B.t.u. of heat, what will be the resulting temperature and pressure?

10. One hundred cubic feet of air at atmospheric pressure (14.5 lb. per sq. in. abs.) and a temperature of 70 deg. fahr. are compressed according to the law $PV^{1.32}$ = constant to a pressure of 87 lb. per sq. in. abs. and then cooled at this pressure to the initial temperature. (*a*) Find the volume at the end of compression. (*b*) What is the volume after cooling at constant pressure? (*c*) Find the total heat removed per pound of air.

11. A cylinder contains 1.5 cu. ft. of a gas at 15 lb. per sq. in. abs. Find the work expended in compressing it to a pressure of 180 lb. per sq. in. abs., the law of compression being $PV^{1.3}$ = constant.

12. The area of an engine piston is 100 sq. in. If the length of the cylinder occupied by a gas is 1.8 in. when the pressure is 120 lb. per sq. in. abs., find the work done by the gas in driving the piston through a distance of 18 in. Take $PV^{1.35}$ = constant.

13. In a gas engine cylinder 5 cu. ft. of a mixture of gas and air at 14.7 lb. per sq. in. abs. are compressed into a space of one cu. ft. If the compression is adiabatic, (*a*) what is the pressure at the end of the compression stroke; and (*b*) how many foot-pounds of work must be expended in the compression of the charge? Take $k = 1.40$.

14. How many foot-pounds of work must be expended in compressing the gas and air mixture of problem 13, if the initial temperature of 100 deg. fahr. is kept constant?

15. If 20 cu. ft. of air are compressed adiabatically from 15 lb. per sq. in. abs., and 60 deg. fahr. to 225 lb. per sq. in. abs., (*a*) find the temperature after the compression, and (*b*) the heat equivalent of the work done.

16. If 15 cu. ft. of air at 62 deg. fahr. and 200 lb. per sq. in. abs. are expanded to a pressure of 25 lb. per sq. in. abs., calculate the final volume and the work done during the expansion, (*a*) if the expansion is isothermal, and (*b*) if the expansion is adiabatic.

17. Given a quantity of air whose volume is 1.3 cu. ft. at a pressure of 15 lb. per sq. in. abs. This air is (1) heated at constant volume until the pressure is 80 lb. per sq. in. abs., (2) expanded adiabatically to a pressure of 15 lb. per sq. in. abs., and (3) compressed at constant pressure to the original volume. (*a*) Draw a PV diagram. (*b*) What is the total heat added in B.t.u.? (*c*) Find the net work done in foot-pounds. (*d*) What is the efficiency of the cycle?

CHAPTER III

STEAM AND STEAM CALORIMETRY

22. Steam Generation.—All materials exist in a solid, liquid, or gaseous state, depending upon their temperature and the pressure exerted upon them. Water under an absolute pressure of 14.7 lb. per sq. in. and at a temperature of 32 deg. fahr. may exist as either a solid or a liquid. Under such conditions its thermal properties are not usually considered in steam engineering.

A sufficiently rapid transfer of heat to water held in an open or closed vessel produces changes in the physical state of the water. The effects are: increased vibration of the molecules, a rise in the temperature, and a decrease in density of the liquid. All of these changes continue until the temperature at which boiling occurs is reached.

At the temperature of ebullition, bubbles of gas form within the water and rise to its surface where they burst. The rapidity with which the bubbles are formed is dependent upon the rate of heat transfer to the water and its movement due to convection currents. The conversion from a liquid into a gaseous state produces vapor. A *vapor* exists as a gas, but it does not follow entirely the laws of a perfect gas. Water, when completely vaporized, is as clear and colorless as clean, dry air. The usual foggy or whitish color, which may be seen in escaping steam, is due to chilling and partial condensation of the steam. The density of the vapor produced is much less than that of the water, while its volume is many times that of the volume of water evaporated. Both the density, or weight per cubic foot, and the volume per pound of steam produced are dependent upon the existing pressure and temperature.

The temperature at which boiling or vaporization occurs is controlled by the absolute pressure exerted upon the water. The boiling temperature is definitely fixed for each pressure, and is less at lower pressures than it is at higher pressures. Water held in an open vessel at an atmospheric pressure of 14.7 lb. per sq. in. abs. will boil rapidly at a temperature of 212 deg. fahr. If the vessel is closed and the pressure increased to 300 lb. per sq. in. abs., the temperature at which vaporization occurs is 417.5 deg. fahr. A reduction of the absolute pressure to

1 in. of mercury within the closed vessel will result in the water boiling at a temperature of 79.06 deg. fahr.

In the production of *saturated* steam the heat added to each pound of water is utilized in two stages. The first stage raises the temperature of the water from that of its initial condition to that which corresponds to the boiling point at the existing absolute pressure. The heat necessary to accomplish this is the *heat of the liquid.* This stage is accomplished by a comparatively small increase of volume. The second stage in the process of steam formation is the addition of an amount of heat necessary to convert the liquid at its boiling point into a vapor. The latter quantity is the *latent heat of vaporization.* The vaporization takes place at a constant temperature corresponding to the boiling point.

In the formation of steam the vapor may be made to have a temperature and heat content higher than that corresponding to the boiling point at its pressure. This third stage is known as *superheating.* The addition of superheat tends to produce a gas that more nearly follows the laws of a perfect gas.

23. Saturated Steam.—Steam which exists at the temperature corresponding to its absolute pressure is defined as *saturated.* Saturated steam may be completely free from unvaporized water particles or it may carry water globules in suspension. Thus, saturated steam may be in either a dry or a wet condition. The term saturated does not indicate a moisture content in the steam, but does, however, indicate that the steam temperature is the normal one corresponding to the absolute pressure.

"The Properties of Dry and Saturated Steam," as determined by Professor G. A. Goodenough, are given in Table II. The values given are based on a weight of 1 lb. of water or steam and the heat contents are measured above a temperature of 32 deg. fahr.

24. Heat of Liquid.—The amount of heat in British thermal units necessary to raise 1 lb. of water from 32 deg. fahr. to the temperature of boiling is *the heat of the liquid.* It should be noted that the heat of the liquid increases as the absolute pressure of the steam increases.

The change of the heat of the liquid per pound of water may be computed as

$$h = c_{pw}(t_s - t_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

where h = heat of the liquid, B.t.u. per lb.
c_{pw} = specific heat of the liquid.
t_s = vaporization temperature, deg. fahr.
t_w = initial temperature of the water, deg. fahr.

TABLE II

PROPERTIES OF DRY AND SATURATED STEAM
(G. A. Goodenough)

Pressure		Temperature	Volume	Heat of	Latent Heat in B.t.u.		Total Heat of	Entropy		
In. of Mercury	Lb. per Sq. In.	Deg. Fahr.	Cu. Ft. per Lb.	Liquid B.t.u.	of Vaporization	Internal	Steam B.t.u.	of Liquid	of Vaporization	of Vapor
	p	t	v	h	L	ρ	H	s'	L/T	s''
0.2	0.0982	34.55	2992.0	2.56	1071.7	1016.3	1074.2	0.0052	2.1687	2.1739
0.5	0.2456	58.83	1255.0	26.91	1058.8	1001.7	1085.7	0.0533	2.0423	2.0956
1.0	0.4912	79.06	652.0	47.11	1047.9	988.7	1095.0	0.0915	1.9455	2.0370
1.5	0.737	91.75	444.5	59.77	1041.1	980.4	1100.8	0.1148	1.8882	2.0030
2.0	0.982	101.17	338.9	69.16	1036.0	974.3	1105.1	0.1316	1.8474	1.9790
2.5	1.228	108.73	274.7	76.70	1031.8	969.4	1108.5	0.1450	1.8155	1.9605
3.0	1.474	115.08	231.4	83.04	1028.3	965.2	1111.4	0.1561	1.7893	1.9454
4.0	1.965	125.44	176.5	93.37	1022.5	958.3	1115.9	0.1739	1.7478	1.9217
5.0	2.456	133.78	143.2	101.68	1017.9	952.8	1119.6	0.1880	1.7154	1.9034
6.0	2.947	140.80	120.7	108.69	1013.9	948.1	1122.6	0.1998	1.6888	1.8886
7.0	3.438	146.88	104.4	114.8	1010.5	944.0	1125.2	0.2098	1.6661	1.8760
8.0	3.929	152.26	92.1	120.2	1007.4	940.4	1127.5	0.2187	1.6464	1.8651
9.0	4.421	157.10	82.5	125.0	1004.6	937.1	1129.6	0.2265	1.6290	1.8556
10.0	4.912	161.5	74.8	129.4	1002.1	934.1	1131.4	0.2336	1.6134	1.8470
11.0	5.403	165.55	68.4	133.4	999.7	931.3	1133.1	0.2401	1.5992	1.8393
12.0	5.894	169.30	63.0	137.2	997.5	928.8	1134.7	0.2461	1.5862	1.8323
13.0	6.39	172.79	58.5	140.7	995.5	926.4	1136.1	0.2516	1.5742	1.8258
14.0	6.88	176.06	54.6	143.9	993.6	924.1	1137.5	0.2568	1.5630	1.8198
15.0	7.37	179.14	51.14	147.0	991.7	922.0	1138.8	0.2617	1.5526	1.8143
20.0	9.82	192.38	39.08	160.3	983.8	912.7	1144.1	0.2822	1.5089	1.7912
25.0	12.28	203.08	31.71	170.1	977.3	905.2	1148.3	0.2986	1.4747	1.7733
28.0	13.75	208.67	28.53	176.6	973.8	901.2	1150.5	0.3070	1.4572	1.7642
29.92	14.697	212.0	26.81	180.0	971.7	898.8	1151.7	0.3120	1.4469	1.7589
30.0	14.74	212.13	26.75	180.1	971.7	898.8	1151.8	0.3122	1.4465	1.7587
	15.0	213.0	26.30	181.0	971.2	898.1	1152.2	0.3135	1.4438	1.7573
	16.0	216.3	24.76	184.3	969.1	895.8	1153.4	0.3184	1.4337	1.7521
	17.0	219.4	23.40	187.5	967.1	893.5	1154.6	0.3230	1.4242	1.7473
	18.0	222.4	22.18	190.5	965.2	891.4	1155.7	0.3274	1.4153	1.7427
	19.0	225.2	21.09	193.3	963.4	889.3	1156.7	0.3316	1.4068	1.7384
	20.0	228.0	20.10	196.0	961.7	887.3	1157.7	0.3356	1.3987	1.7343
	30.0	250.3	13.76	218.6	947.1	870.7	1165.7	0.3679	1.3340	1.7019
	40.0	267.2	10.51	235.8	935.5	857.8	1171.3	0.3917	1.2871	1.6788
	50.0	281.0	8.53	249.8	925.9	847.1	1175.6	0.4108	1.2501	1.6609
	60.0	292.7	7.18	261.7	917.4	837.8	1179.1	0.4267	1.2195	1.6462
	70.0	302.9	6.22	272.2	909.8	829.5	1182.0	0.4405	1.1931	1.6336
	80.0	312.0	5.48	281.6	902.8	821.9	1184.4	0.4527	1.1700	1.6227
	90.0	320.3	4.905	290.1	896.4	815.0	1186.5	0.4636	1.1495	1.6131
	100.0	327.8	4.442	297.9	890.5	808.6	1188.4	0.4736	1.1309	1.6045
	105.0	331.4	4.240	301.6	887.6	805.5	1189.2	0.4782	1.1222	1.6004
	110.0	334.8	4.057	305.1	884.8	802.6	1190.0	0.4827	1.1138	1.5965
	115.0	338.1	3.889	308.6	882.1	799.7	1190.7	0.4870	1.1058	1.5928
	120.0	341.3	3.735	311.9	879.5	796.9	1191.4	0.4911	1.0982	1.5893
	125.0	344.4	3.593	315.1	876.9	794.2	1192.0	0.4950	1.0908	1.5858
	130.0	347.4	3.461	318.2	874.4	791.6	1192.6	0.4989	1.0836	1.5825
	135.0	350.3	3.340	321.2	872.0	789.0	1193.2	0.5026	1.0767	1.5793

TABLE II—*Continued*

PROPERTIES OF DRY AND SATURATED STEAM
(G. A. Goodenough)

Pressure Lb. per Sq. In.	Temperature Deg. Fahr.	Volume Cu. Ft. per Lb.	Heat of Liquid B.t.u.	Latent Heat in B.t.u.		Total Heat of Steam B.t.u.	Entropy		
				of Vaporization	Internal		of Liquid	of Vaporization	of Vapor
p	t	v	h	L	ρ	H	s'	L/T	s''
140	353.1	3.226	324.2	869.6	786.4	1193.7	0.5062	1.0700	1.5762
145	355.8	3.120	327.0	867.2	784.0	1194.2	0.5097	1.0636	1.5733
150	358.5	3.020	329.8	864.9	781.6	1194.7	0.5131	1.0573	1.5704
155	361.1	2.927	332.5	862.7	779.2	1195.2	0.5164	1.0512	1.5676
160	363.6	2.839	335.2	860.5	776.9	1195.7	0.5196	1.0453	1.5649
165	366.1	2.757	337.8	858.3	774.6	1196.1	0.5227	1.0395	1.5622
170	368.5	2.679	340.3	856.2	772.4	1196.5	0.5258	1.0339	1.5597
175	370.8	2.605	342.8	854.1	770.2	1196.9	0.5287	1.0284	1.5572
180	373.1	2.536	345.2	852.0	768.0	1197.2	0.5316	1.0231	1.5547
185	375.4	2.470	347.6	849.9	765.9	1197.6	0.5344	1.0179	1.5523
190	377.6	2.408	350.0	847.9	763.9	1197.9	0.5372	1.0128	1.5500
195	379.7	2.348	352.2	846.0	761.8	1198.2	0.5399	1.0079	1.5478
200	381.9	2.292	354.5	844.0	759.8	1198.5	0.5426	1.0030	1.5456
210	386.0	2.186	358.8	840.2	755.9	1199.0	0.5477	0.9936	1.5413
220	390.0	2.090	363.0	836.5	752.1	1199.5	0.5526	0.9846	1.5372
230	393.8	2.002	367.1	832.8	748.3	1199.9	0.5573	0.9760	1.5333
240	397.5	1.921	371.0	829.3	744.7	1200.3	0.5619	0.9676	1.5295
250	401.1	1.846	374.9	825.8	741.2	1200.6	0.5663	0.9595	1.5258
260	404.5	1.777	378.6	822.4	737.7	1201.0	0.5706	0.9517	1.5223
270	407.9	1.713	382.2	819.1	734.4	1201.2	0.5747	0.9442	1.5189
280	411.2	1.654	385.7	815.8	731.1	1201.5	0.5787	0.9369	1.5156
290	414.4	1.598	389.1	812.6	727.9	1201.7	0.5826	0.9298	1.5123
300	417.5	1.545	392.4	809.4	724.7	1201.9	0.5863	0.9229	1.5092
310	420.5	1.496	395.7	806.4	721.6	1202.0	0.5900	0.9162	1.5062
320	423.4	1.450	398.9	803.3	718.5	1202.2	0.5935	0.9097	1.5032
330	426.3	1.407	402.0	800.3	715.6	1202.3	0.5970	0.9034	1.5004
340	429.1	1.366	405.0	797.4	712.6	1202.4	0.6004	0.8972	1.4976
350	431.9	1.327	408.0	794.5	709.7	1202.5	0.6036	0.8912	1.4949
375	438.5	1.239	415.1	787.5	702.7	1202.6	0.6115	0.8768	1.4884
400	444.8	1.162	422.0	780.6	695.9	1202.5	0.6190	0.8631	1.4821
425	450.7	1.094	428.5	773.9	689.4	1202.4	0.6261	0.8501	1.4762
450	456.5	1.033	434.8	767.4	683.1	1202.2	0.6329	0.8377	1.4706
475	462.0	0.978	440.8	761.1	676.8	1201.9	0.6393	0.8260	1.4653
500	467.2	0.928	446.6	755.0	670.9	1201.7	0.6455	0.8146	1.4601
550	477.2	0.843	457.6	743.2	659.4	1200.8	0.6571	0.7934	1.4505
600	486.5	0.770	468.0	731.8	648.5	1199.8	0.6679	0.7735	1.4414
650	495.2	0.708	477.8	720.9	638.0	1198.7	0.6780	0.7550	1.4330
700	503.4	0.656	487.1	710.3	627.9	1197.4	0.6874	0.7376	1.4250
750	511.1	0.610	495.9	700.0	618.2	1195.9	0.6963	0.7212	1.4175
800	518.5	0.570	504.3	690.1	608.8	1194.4	0.7048	0.7056	1.4104
850	525.5	0.534	512.5	680.4	599.7	1192.8	0.7128	0.6907	1.4035
900	532.3	0.502	520.3	670.8	590.8	1191.1	0.7205	0.6764	1.3969
1000	544.9	0.447	535.2	652.4	573.6	1187.6	0.7349	0.6496	1.3845
1100	556.6	0.403	549.1	634.7	557.3	1183.8	0.7482	0.6247	1.3729
1200	567.7	0.364	562.3	617.6	541.8	1179.7	0.7607	0.6015	1.3622

The specific heat of water varies slightly between 32 and 212 deg. fahr. If exact results are desired, the heats of the liquids as given in column 5 of Table II should be used. For many calculations results of sufficient accuracy may be obtained by considering the specific heats as unity for temperatures less then 212 deg. fahr. Equation 47 may then be written

$$h = (t_s - 32) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

25. Latent Heat of Vaporization.—The heat in British thermal units necessary to convert 1 lb. of liquid into dry steam at the same temperature and pressure is *the total latent heat of vaporization* L. It should be noted, by examination of the total latent heats of vaporization L given in Table II, that the amount of latent heat required for vaporization decreases as the pressure increases.

The total latent heat of vaporization is made up of two parts, the *external* and the *internal* latent heats. The external latent heat APv is the heat equivalent of the external work necessary to change the volume of one pound of water to the volume occupied by one pound of the dry steam formed at the existing pressure. The external work equivalent in B.t.u. is expressed as

$$APv = \frac{144p(v_s - v_w)}{778} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

APv = external latent heat of vaporization, B.t.u. per lb.
p = steam pressure, lb. per sq. in. abs.
v_s = volume of steam formed, cu. ft. per lb.
v_w = volume of water at vaporization temperature, cu. ft. per lb.

The internal latent heat of vaporization ρ is the heat equivalent, in B.t.u., of the internal work necessary to overcome the molecular attraction of the water molecules at the vaporization temperature. The internal latent heat of vaporization is equal to the total latent heat L minus the external latent heat APv.

$$\rho = (L - APv) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

The internal latent heats are given in Table II and the external latent heats may be easily obtained from the table values as

$$APv = (L - \rho) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

26. Total Heat of Dry Saturated Steam.—The total heat of one pound of dry saturated steam is the sum of the heat of the liquid and the total latent heat of vaporization.

$$H = L + h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

H = total heat above 32 deg. fahr., B.t.u. per lb.
L = total latent heat of vaporization, B.t.u. per lb.
h = heat of the liquid, B.t.u. per lb.

When use is made of equation 52, the values of L and h are taken from Table II for the absolute steam pressure corresponding to that of the steam under consideration.

27. Quality of Steam.—Saturated steam may or may not carry entrained water. Moisture present in steam may be due to several causes. Steam conveyed by piping may suffer a reduction in heat content due to unavoidable or other losses, so that a part of the latent heat of the steam is given up and water particles are formed in the steam. The generation of the steam may have been imperfect due to poor design of the boiler, high rate of evaporation maintained, or poor boiler feedwater.

The dry steam content per pound of wet steam is the *quality* of the steam. The quality x of steam may be expressed as a percentage, or it may be stated as the weight of dry steam existing within a pound of wet steam.

28. Total Heat of Wet Saturated Steam.—The total heat of a pound of wet saturated steam is less than that of a pound of dry saturated steam under the same pressure conditions. When the water is not completely vaporized the heat carried as the heat of vaporization is less. The quality affects only the latent heat of vaporization and does not change the heat of the liquid at any given pressure. The quality factor must be applied only to the latent heat of vaporization; and not to the heat of the liquid or the total heat of a pound of dry saturated steam. The total heat of a pound of wet steam is

$$H = xL + h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

H = total heat of wet steam, B.t.u. per lb.
x = quality of the steam.
L = total latent heat of vaporization, B.t.u. per lb.
h = heat of the liquid, B.t.u. per lb.

29. Superheated Steam.—Saturated steam, at any pressure, may have its temperature and total heat content increased by the application of additional heat. When the steam temperature is above the

saturation temperature, corresponding to the steam pressure, the steam is superheated. It is possible to have superheated steam existing in contact with the water from which it was formed. However, the foregoing condition is not commercially feasible; therefore the superheating process usually takes place in a separate piece of equipment where the steam is not in contact with its liquid.

The total heat content of a pound of superheated steam is made up of three items: (1) the heat of the liquid, (2) the latent heat of vaporization, and (3) the heat added as superheat. In equation form the foregoing is expressed as

$$H = L + h + c_p(t_{sh} - t_s) \quad . \quad . \quad . \quad . \quad . \quad (54)$$

H = total heat of superheated steam, B.t.u. per lb.
L = latent heat of vaporization, B.t.u. per lb.
h = heat of liquid, B.t.u. per lb.
c_p = mean specific heat of superheated steam at the temperature and pressure of superheat.
t_{sh} = temperature of the superheated steam, deg. fahr.
t_s = temperature of saturated steam at the existing pressure, deg. fahr.

The specific heat of superheated steam varies with the existent pressure and the amount of the superheating. The temperature difference $(t_{sh} - t_s)$ is the number of degrees of superheat. It is indicative of the rise of the temperature above that of saturation conditions. Unless the mean specific heat of the superheated steam is known, the heat added as superheat cannot be computed from the temperature rise of the steam. Table III gives values of the mean specific heats of superheated steam computed from the "Properties of Superheated Steam," as given by Professor G. A. Goodenough. The values given in Table III are for various absolute pressures and number of degrees fahrenheit of superheat.

The advantages of superheating steam are: (1) the additional heat imparted to the vapor causes it to behave more nearly as a perfect gas; (2) the temperature ranges in certain pieces of steam-using equipment may be increased; and (3) steam condensation losses may be eliminated or lessened in pipe lines and steam-using engines and turbines.

30. Specific Volume and Density of Dry Saturated Steam.—The specific volumes v, in cubic feet, occupied by one pound of dry saturated steam are given in column 4 of Table II. The volume of one pound of steam varies inversely as the pressure, and at low pressures the volumes per pound are very much greater than they are per pound of steam at higher pressures.

TABLE III

MEAN SPECIFIC HEATS OF SUPERHEATED STEAM

Steam Pressure, Lb. per Sq. In. Abs.	Superheat, Degrees Fahrenheit									
	10	20	40	60	80	100	150	200	250	300
10	0.484	0.482	0.478	0.475	0.474	0.473	0.472	0.470	0.470	0.470
15	0.490	0.488	0.485	0.482	0.481	0.480	0.475	0.473	0.472	0.470
20	0.498	0.496	0.493	0.490	0.488	0.485	0.480	0.478	0.477	0.475
50	0.521	0.519	0.516	0.513	0.510	0.508	0.503	0.500	0.497	0.495
100	0.563	0.560	0.554	0.548	0.543	0.540	0.530	0.525	0.520	0.516
150	0.603	0.598	0.590	0.582	0.575	0.569	0.558	0.549	0.543	0.537
200	0.630	0.625	0.617	0.608	0.600	0.593	0.580	0.569	0.561	0.555
250	0.660	0.655	0.645	0.636	0.628	0.620	0.602	0.590	0.580	0.572
300	0.678	0.674	0.665	0.657	0.648	0.640	0.620	0.607	0.597	0.587
350	0.703	0.698	0.688	0.678	0.668	0.659	0.638	0.624	0.612	0.601
400	0.732	0.725	0.715	0.704	0.693	0.683	0.660	0.642	0.627	0.615
500	0.772	0.765	0.752	0.740	0.728	0.718	0.693	0.673	0.656	0.642
600	0.817	0.809	0.793	0.778	0.765	0.753	0.724	0.700	0.682	0.667
700	0.847	0.840	0.826	0.812	0.798	0.785	0.754	0.728	0.709	0.692
800	0.910	0.890	0.860	0.842	0.828	0.814	0.782	0.755	0.733	0.713

The density d of dry saturated steam is the weight of steam in pounds per cubic foot. The density is the reciprocal of the specific volume per pound, and is

$$d = \frac{1}{v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

The density varies as the pressure; thus the higher the pressure the greater the weight of the steam per cubic foot.

31. Steam Quality Determination.—The dryness or quality of saturated steam is determined by the use of some type of a steam calorimeter. If the quality of the steam x is known, the moisture content in per cent is equal to $(100 - x)$.

Two types of steam calorimeters for the determination of either steam quality or the moisture content are in use. These are the *separating* calorimeter, and the *throttling* or *superheating* calorimeter. Combinations of these types may be employed where the steam is of low quality, and where a greater degree of accuracy is required.

When a quality or moisture determination is made, it is imperative that a representative sample of the steam be passed through the calorim-

eter. A typical sample of the steam can be secured only by the use of a properly designed and located sampling nozzle. The sampling nozzle recommended by the American Society of Mechanical Engineers consists of a short piece of $\frac{1}{2}$-in. pipe closed at one end. The closed end of the pipe should be inserted into the steam supply line so as to extend nearly across the main. The steam enters such a nozzle through $\frac{1}{8}$-in. holes drilled along a spiral on the cylindrical surface of the pipe. None of the $\frac{1}{8}$-in. holes should be closer than $\frac{1}{2}$-in. from the walls of the steam main. The sampling nozzle should be located preferably in a vertical main as far as possible from pipe bends and fittings. The precautions are necessary to insure the withdrawal of a sample, in which the entrained moisture is not unduly increased by condensed steam flowing along the walls of the steam main. The connection between the sampling nozzle and the calorimeter should be as short as possible. Both the calorimeter and the calorimeter connections should be well insulated.

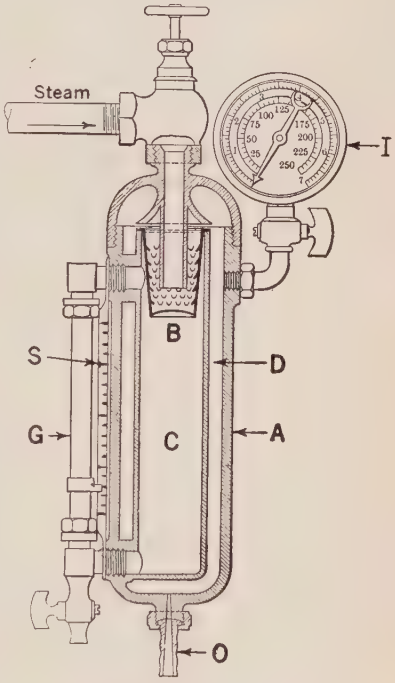


Fig. 11.—Separating calorimeter.

32. Separating Calorimeter.—This is a device for mechanically separating the entrained moisture from the steam and collecting it so that the moisture may be accurately measured. Provision must also be made for the measurement of the amount of dried steam escaping at the calorimeter exhaust. The construction of a typical separating calorimeter is illustrated by Fig. 11.

The calorimeter body A has an inner chamber C at the top of which is placed the separating baffle B. The incoming steam strikes the perforated baffle and is made to reverse its direction of flow. The reversal of the steam flow, together with the impingement upon the mesh, causes the water particles to be separated and dropped to the bottom of the inner chamber. The level of the moisture collected in the inner chamber may be observed in the gage glass G connected with the chamber. A graduated scale S is held along the gage glass; and from this scale the weight of the moisture collected within the calorimeter is read. The

smallest graduation on this scale is usually 0.01 lb.; by careful estimation the weight of moisture may be read to 0.001 lb.

The dry steam escapes from the perforated baffle or cup through an annular chamber D which surrounds the inner chamber. The outer chamber serves as a steam jacket which reduces the radiation and other heat losses from the incoming steam. The steam escapes from the outer chamber through an orifice O located at the bottom of the calorimeter body. The steam may be discharged to the air or into a condenser. The latter is preferable for the accurate determination of the weight of dry steam escaping during a given period of time, as the dry steam condensed may be weighed. A pressure gage I is attached to the outer chamber so that the pressure within the calorimeter may be determined. This gage also has graduations upon its dial indicating the flow of dry steam through the orifice, per ten minutes of time. The weight of dry steam flowing per second may also be calculated by Napier's rule when the discharge pressure is less than 0.6 of the initial pressure,

$$w = \frac{pa}{70} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

w = weight of dry steam per second, lb.
p = steam pressure, lb. per sq. in. abs.
a = area of the orifice, sq. in.

The attached pressure and flow gage is subject to many errors and may give erroneous readings. Also, the orifice may become clogged with dirt and rust, or it may be corroded so that even though the gage is mechanically correct the readings may be misleading. Likewise, the calculations of the weight of steam by Napier's rule may be unreliable, if the diameter of the orifice is not accurately known or if the orifice is partially clogged.

The steam quality in per cent is given by the equation,

$$x = \frac{W_d \times 100}{W_d + W_m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

x = quality of the steam, per cent.
W_d = weight of dry steam passing through the calorimeter, lb.
W_m = weight of moisture separated from W_d lb. of dry steam, lb.

Separating calorimeters should be used for the determination of the quality of low pressure steam, or very wet steam.

33. Throttling Calorimeters.—All calorimeters of this type operate upon the principle that if steam is expanded without doing any

external work, or without the loss of heat, the total energy of the steam is unchanged. Radiation losses should be minimized by having the calorimeter, as well as its connection to the steam main, thoroughly insulated. The expansion of the steam is accomplished by allowing the steam to flow through a small orifice from a region of high pressure to one of low pressure.

A throttling calorimeter, Fig. 12, can be constructed of pipe fittings and a thin metal disk. The disk should have a $\frac{1}{8}$-in. diameter orifice opening centrally located. All burrs or roughness should be removed from the disk about the orifice opening. The essential fittings required

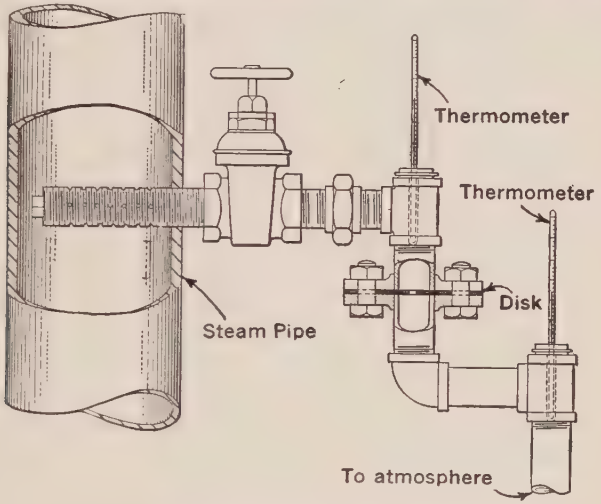


Fig. 12.—Throttling calorimeter and sampling nozzle.

are two tees, into which thermometer wells are inserted, a pair of pipe flanges, and connecting pipe nipples and elbows. The disk is held between the two flanges which are located as shown. A good gasket should be inserted on each side of the orifice plate to prevent steam leakage. The calorimeter is attached to the standard sampling nozzle with a shut-off valve between the calorimeter and the nozzle. An accurate steam pressure gage connected on the supply side of the orifice is desirable, but is not absolutely necessary. The steam temperature reading taken on this side may be used to secure the same results when the steam is not initially superheated. The discharge from the calorimeter is direct to the air at atmospheric pressure.

Another form of throttling calorimeter is illustrated by Fig. 13. This calorimeter consists of a chamber with a naked thermometer, or a thermometer well, centrally placed within it. Steam enters the calorimeter through a small orifice located in one side of the body. A mercury

manometer is attached to the calorimeter shell and is used to measure the pressure of the steam within the thermometer chamber. The calorimeter discharge may be directly to the air or through suitable piping to any convenient place. The function of the Bourdon gage is to give the pressure of the steam flowing to the calorimeter.

The operation of either of the two throttling calorimeters is accomplished by fully opening the valve in the calorimeter connection and allowing the steam to expand through the orifice. If no work is done and there are no radiation losses, the total heat content of the steam is unchanged during the process. However, the total heat content of the high-pressure wet steam is greater than that normal for saturated steam at the absolute pressure after expansion. This excess of heat is utilized in evaporating the moisture of the steam and superheating the steam at the lower pressure. The temperature of the low-pressure steam is measured by the thermometer inserted into the calorimeter body. The relation may be established: the total heat of the wet saturated steam before expansion, $x_1L_1 + h_1$, is equal to the total heat of the superheated steam after expansion, or $H_2 + c_p(t_c - t_s)$.

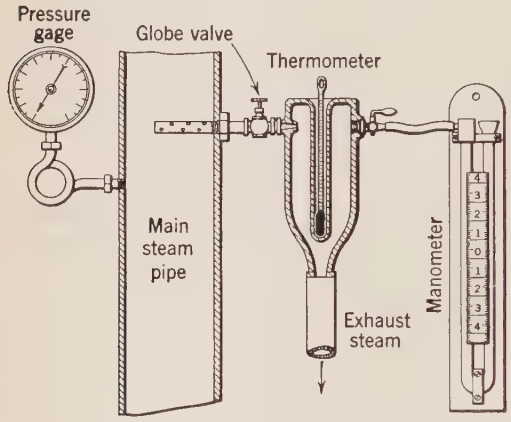


FIG. 13.—Carpenter throttling calorimeter.

$$x_1L_1 + h_1 = H_2 + c_p(t_c - t_s) \quad . \quad . \quad . \quad . \quad . \quad (58)$$

x_1 = initial quality of the steam.

L_1 = total latent heat of vaporization at the initial pressure abs., B.t.u. per lb.

h_1 = heat of the liquid at the initial pressure abs., B.t.u. per lb.

H_2 = total heat of dry saturated steam at the final pressure abs., B.t.u. per lb.

c_p = mean specific heat of superheated steam.

t_c = temperature of the superheated steam, deg. fahr.

t_s = temperature of saturated steam corresponding to the final absolute pressure in the calorimeter, deg. fahr.

$$x_1 = \frac{H_2 + c_p(t_c - t_s) - h_1}{L_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

Throttling calorimeters should be well insulated and at least 10 deg. fahr. of superheat must be obtained by the expansion of the steam. Throttling calorimeters exhausting to the atmosphere may not be expected to give accurate quality determinations with steam having moisture contents greater than those listed for the following gage pressures:

Gage Pressure, Lb. Sq. In.	Maximum Moisture Content, Per Cent
40	2.5
150	4.5
250	5.5
600	6.0

34. Calorimeter Errors.—One of the chief errors in steam quality determinations is caused by the inability to secure a composite sample of the steam. The following are some of the errors that may occur in the use of calorimeters:

Separating.—Condensation between the nozzle and the calorimeter, incomplete separation of the moisture, inaccuracies of the steam-flow gage, or outlet orifice clogged or corroded rendering incorrect the flow gage readings or the calculations by Napier's rule.

Throttling.—Radiation losses, pressure gage errors, orifice opening clogged with dirt, thermometers in wells do not respond readily to temperature variations, stem corrections not applied to the exposed thermometer stems, or errors in the specific heats of superheated steam.

35. Change of Entropy.—The ratio of the amount of heat added to or taken from a pound of steam to the average absolute temperature at which the addition or the abstraction is made expresses the *change of entropy*. Thus, if the total latent heat of vaporization L is added to one pound of water at its boiling temperature T, deg. fahr. abs., and the vaporization is complete, the ratio or the change in entropy at vaporization is L/T. The entropy change is the change in the heat content, per pound, per degree of absolute temperature.

The total entropy s'' of saturated steam is comprised of two parts, the entropy of the liquid s', and the entropy of vaporization L/T. Numerical values for these items for various absolute pressures may be had by reference to Table II. The change of entropy of the liquid cannot be expressed so simply as the change of entropy of vaporization, because of the rise in temperature and the variation in the specific heat of water between 32 deg. fahr. and the temperature corresponding to the pressure. In the case of the liquid, if the absolute temperatures of the liquid be plotted as ordinates with the corresponding entropy changes as abscissas, the resulting temperature-entropy diagram will be a curve,

Fig. 14. The average absolute temperature is not, therefore, the numerical mean of the initial and final temperatures. It is necessary to consider that the heat has been added in infinitely small amounts at the changing absolute temperatures, between the initial and the final conditions. When temperature changes occur the condition is as shown by Fig. 14.

For any given temperature T, $ds' = \dfrac{dh}{T}$ and the area of the strip dh is equal to $T\,ds'$. The summation of the infinitely narrow strips between s'_1 and s'_2 or the change in entropy is

$$s'_2 - s'_1 = \int_{T_1}^{T_2} c_{pw} \frac{dT}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

When the specific heat $c_{pw} = 1$, the expression when integrated becomes

$$s'_2 - s'_1 = \log_e T_2 - \log_e T_1 = \log_e \frac{T_2}{T_1} \quad . \quad . \quad . \quad (61)$$

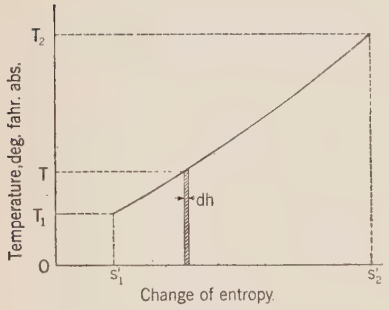


Fig. 14.—Temperature-entropy diagram, liquid.

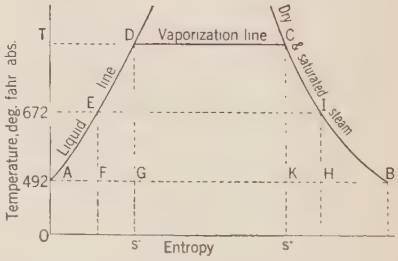


Fig. 15.—Temperature-entropy diagram, dry saturated steam.

Fig. 15 is a temperature-total entropy diagram for one pound of water converted into dry saturated steam at T deg. fahr. abs. An examination of the numerical quantities given as the entropy changes at the various pressures and temperatures of "Properties of Dry Saturated Steam," Table II, reveals three things. When the steam pressure and temperature are increased the entropy change of the liquid increases, the entropy change at vaporization decreases, and the total entropy change incurred is decreased. In Fig. 15 the path of the plotted values of the entropy changes in the liquid for intermediate temperatures fall along the liquid line AD, while those for the entropy of vaporization fall along the dry and saturated steam line BC. Thus, at a temperature of 492 deg. fahr. abs. the total entropy change is represented by the length of the line AB. In this particular case the entropy change of the

liquid is zero and the entropy change at vaporization is equal to AB. In the case of water evaporated at 212 deg., or 672 deg. fahr. abs., the entropy change of the liquid is represented by the line AF and the entropy of vaporization by the line EI, so that the total change of entropy is depicted by the line AH. When the vaporization occurs at a temperature T the entropy changes are represented by the lines AG and DC. It should be noted that if the pressure and the temperature be made high enough, the curves AD and BC will intersect. Such a condition would indicate that at some pressure water would pass from a liquid to a vapor without the addition of any latent heat of vaporization; and that the total heat of the steam would be comprised of the heat of the liquid only.

36. Entropy Changes of Wet Steam.—When steam carries moisture in suspension, the total entropy changes and those at vaporization are less than those indicated for the pressures of Table II. In the calculation of the heat content of a pound of wet saturated steam the quality factor x affects only the total latent heat of vaporization L. At any given pressure the heat of the liquid is never changed by a quality factor. Hence, in the calculation of the change of entropy of a pound of water converted into wet steam, the entropy of vaporization is multiplied by the quality. The total entropy change is expressed

$$s''_1 = s' + x\frac{L}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

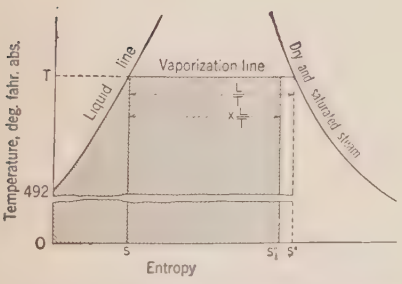


Fig. 16.—Temperature-entropy diagram, wet steam.

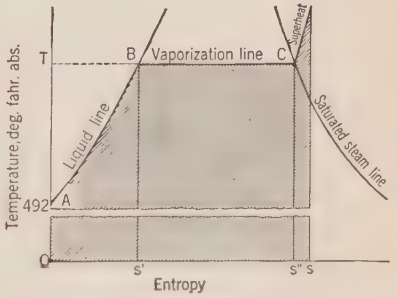


Fig. 17.—Temperature-entropy diagram, superheated steam.

A graphical illustration of the use of equation 62 is given by Fig. 16.

37. Entropy Changes of Superheated Steam.—The addition of a sufficient amount of heat to saturated steam without an increase in the steam pressure produces superheated steam. The steam temperature and the total entropy change are increased. Fig. 17 illustrates the temperature-entropy changes of superheated steam. The change of entropy

due to superheating is delineated by the curve at C sloping upward to the right. Entropy changes during superheating occur with increasing temperatures and varying specific heats. Consequently the method of calculating the changes, that occur during the superheating process, is similar to that employed for entropy variations of a liquid. The student is referred to steam tables giving the entropy changes for superheated steam.

38. Constant Entropy.—When adiabatic conditions are maintained the heat of a material is neither given to nor received from external bodies. Under adiabatic conditions there is no entropy change. However, if steam expands adiabatically from a condition of higher temperature to one of lower temperature and work is done, there is a reduction in the heat content of the steam. If the expansion of superheated steam is carried far enough, or if the steam is initially saturated, the quality of the steam after expansion will be less than 100 per cent. If the initial steam conditions are known and the final steam pressure after adiabatic expansion is known, the final quality may be computed. With no change of entropy during adiabatic expansion in which work is done, the initial total entropy is equal to the final total entropy.

$$s'_1 + x_1\frac{L_1}{T_1} = s'_2 + x_2\frac{L_2}{T_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

$$x_2 = (s'_1 + x_1\frac{L_1}{T_1} - s'_2)\frac{T_2}{L_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

The numerical quantities are to be taken directly from the steam tables.

39. Total Heat-Entropy Diagrams.—The Mollier diagram, Fig. 18, affords a ready and rapid aid in solving problems involving the properties of steam. From this chart the total heat and entropy changes of steam at various pressures may be read for either a dry, wet, or superheated condition. The use of the diagram permits a quick determination of the quality of steam from data given by a throttling calorimeter test. The total heat and quality of steam may be determined after adiabatic expansion, also from the chart, when the initial steam conditions are known.

Example.—Find the total heat of 1 lb. of steam at an absolute pressure of 160 lb. per sq. in. abs. and 97 per cent quality.

Solution.—Locate on the chart of Fig. 18 the intersection of the 97 per cent quality line and the 160 lb. per sq. in. pressure line. Read along a horizontal line, either to the right or the left, the total heat of the steam, 1170 B.t.u. per lb.

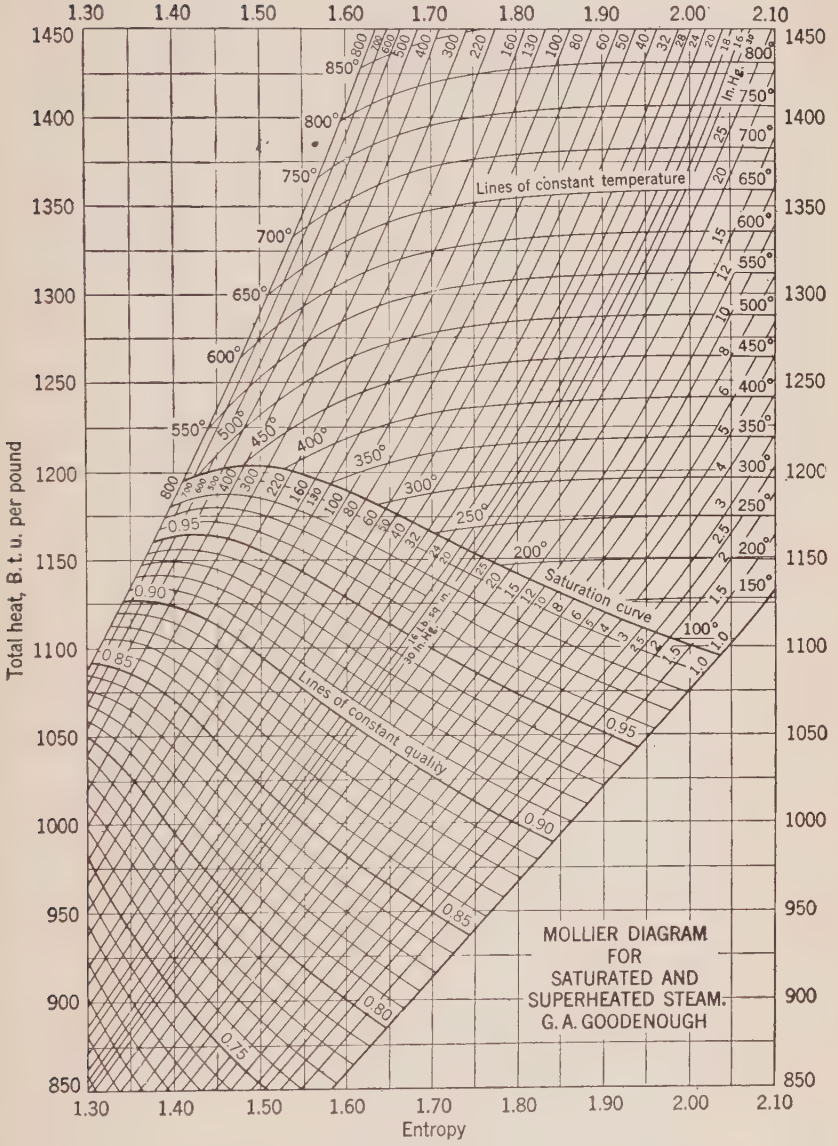


Fig. 18.—Total heat-entropy diagram, saturated and superheated steam.

Example.—Steam at a pressure of 160 lb. per sq. in. abs. had a temperature of 550 deg. fahr. Find the total heat per pound.

Solution.—Find the intersection of the 160 lb. per sq. in. pressure line and the 550 deg. fahr. temperature line. The total heat corresponding to this intersection is 1299 B.t.u. per lb.

Example.—A throttling calorimeter test gave the following data: average initial steam pressure 130 lb. per sq. in. abs., calorimeter exhaust pressure 16 lb. per sq. in. abs., temperature of the steam in the calorimeter 250 deg. fahr. Find the quality of the steam.

Solution.—Locate the intersection of the 250-deg. line and the 16 lb. per sq. in. pressure line. Move horizontally from this point to the 130 lb. per sq. in. pressure line, remembering that the total heat is unchanged. At the intersection of the horizontal line and the 130-lb. line read the quality 0.975.

Example.—Steam at 220 lb. per sq. in. abs. and 0.99 quality is expanded adiabatically to a pressure of 5 in. of mercury absolute. Find the total heat and the quality of the steam after expansion.

Solution.—From the intersection of the 220 lb. per sq. in. pressure line and the 0.99 quality line drop vertically along a line of constant entropy to the 5 in. of mercury pressure line. There a quality of 0.781 and a total heat of 897 B.t.u. per lb. may be read.

PROBLEMS

1. Find the temperatures at which vaporization occurs for the following pressures, lb. per sq. in. abs.; 50, 70, 135, 200, 370, and 1150.

2. Find the heats of the liquid at the following absolute pressures; 100, 147, 198, 450, and 610.

3. One pound of water has a temperature of 100 deg. fahr. Assume the specific heat of the water as 1. What is the heat of the liquid?

4. Two pounds of water exist at a temperature of 353.1 deg. fahr. Mean specific heat of the water 1.01. What is the heat of the liquid?

5. Forty pounds of water have a temperature rise from 250.3 deg. fahr. to 401.1 deg. fahr. What was the increase in the heat of the liquid? What absolute pressures are represented by vaporization temperatures of 250.3 and 401.1 deg. fahr.?

6. From Table II find the total latent heats of vaporization for the pressures given in problem 2.

7. One cubic foot of water at 417.5 deg. weighs 52.66 lb. If steam is formed at 300 lb. per sq. in. abs., the volume of the steam per pound is 1.545 cu. ft. Calculate the external latent heat of vaporization per pound under the above conditions.

8. What is the total latent heat of vaporization for one cubic foot of water under the conditions of problem 7? What is the internal latent heat of vaporization per pound?

9. Find the total heats of dry saturated steam per pound at the following gage pressures; 125.5, 175, 250, 325, and 400. Barometer 29.6 in. of mercury.

10. Find the total heat content of 20,000 lb. of dry saturated steam at 225 lb. per sq. in. gage. Barometer 29.53 in. Hg.

11. Twenty pounds of water at a temperature of 80 deg. fahr. were converted

into dry saturated steam at a pressure of 169 lb. per sq. in. gage. Barometer 29.7 in. Hg. How much heat was added to the water?

12. Find the total heat content of wet saturated steam the quality of which is 98.5 per cent at the following gage pressures; 60, 90, 157, 324, and 410 lb. per sq. in. Barometer 29.8 in. Hg.

13. Find the densities of dry saturated steam at the following absolute pressures; 95, 165, 275, 360, and 1200 lb. per sq. in.

14. How much heat would have been required for the conditions of problem 11, had the steam formed had a quality of 97 per cent?

15. Find the total heat content of one pound of steam with 100 deg. fahr. of superheat at the following absolute pressures: 150, 200, 300, 350, and 450 lb. per sq. in.

16. A boiler equipped with a superheater was supplied with water at a temperature of 176 deg. fahr. The steam pressure carried was 635.5 lb. per sq. in. gage and the temperature of the steam leaving the superheater was 645.2 deg. fahr. Barometer 29.6 in. Hg. How many degrees of superheat existed? How much heat was added per pound of water to convert it into superheated steam?

17. A separating calorimeter with a 0.08-in.-diameter orifice gave the following test data: steam pressure 145 lb. per sq. in. gage, moisture collected per test 0.35 lb., dry steam condensed per test of 10 minutes 6.8 lb. Barometer 29.6 in. Hg. What was the steam quality?

18. If the calorimeter flow gage indicated a flow of 8.1 lb. of dry steam per 10 minutes during the test mentioned in problem 17, what was the percentage of error? Assume that the correct quality was obtained by the method of problem 17.

19. Calculate the steam quality in problem 17, basing the dry steam discharge on Napier's rule.

20. A throttling calorimeter gave the following test data: steam pressure in the main 145 lb. per sq. in. gage, barometer 29.92 in. Hg, calorimeter exhaust to the atmosphere, temperature of the steam in the calorimeter 265 deg. fahr. What was the quality of the steam?

21. What would have been the quality had the calorimeter of problem 20 exhausted against a pressure of 5 lb. per sq. in. gage, other conditions of pressure and temperature being the same?

22. What would have been indicated if the calorimeter temperature of problem 20 had been 212 deg. fahr.? 350 deg. fahr.?

23. Check by calculation the change of entropy of the liquid of water from 32 deg. fahr. to 212 deg. fahr.

24. Calculate the entropy change at vaporization for the following absolute pressures: 120, 185, 275, 360, and 550. Find the total entropy changes for the same pressures.

25. One pound of wet steam of 98.5 per cent quality is expanded adiabatically from 250 lb. per sq. in. gage to 2 lb. per sq. in. gage. Barometer 29.5 in. Hg. Calculate the quality of the steam after expansion. Check by use of the Mollier diagram.

End 10/3,

CHAPTER IV

FUELS

40. Production of Heat.—In commercial power plants the heat energy used to produce steam or mechanical energy is released from a fuel. This heat energy is secured by the rapid oxidation of some material which may be either in its natural state or in a prepared form. The only requirements are that the fuel be anything that can be burned without great difficulty, and that sufficient quantities are available at prices not prohibitive.

41. Forms of Fuels.—The common fuels used in the production of heat for commercial purposes are in either a *solid*, a *liquid*, or *a gaseous state*. Each of these three fuel forms may be classified according to the preparation they have undergone:

(1)	Solid	Natural: coal, lignites, and wood wastes. Prepared: coke.
(2)	Liquid	Natural: crude petroleum oils. Prepared: petroleum distillates.
(3)	Gaseous	Natural: natural gas. Prepared: blast-furnace gas, producer gas, coke-oven gas, coal gas, etc.

Some form of coal is the most commonly used fuel. Wood is too expensive and bulky for the amount of heat contained to have an extensive use. In manufacturing plants where wood wastes are by-products, these may be used for the production of heat with economical results.

Fuel oil has an extensive field of use on shipboard and near oil fields or oil distributing centers. Due to improved methods of refining crude petroleums, in the production of gasoline, many distillates formerly used by power plants are not now available. Heavy oils, oil distillates of various grades, and gasoline are extensively used as fuels for internal-combustion engines.

Gaseous fuels of all kinds are suitable for internal-combustion engines. The use of natural gas for steam power is small and is limited to localities

where the gas is produced. Only in the case of the iron and steel industry is any considerable quantity of steam produced for power purposes by the use of by-product gases. In many cases the maintenance costs of internal-combustion engines have proved to be large, consequently there is a marked tendency in the metallurgical industries to employ electrical motor drives. The electrical power under such conditions is supplied by steam turbine-driven generators. With modern steam turbines and condensing equipment good economy can be obtained with lowered maintenance costs. By-product coke-oven and blast-furnace gases are burned under boilers for the production of steam for the plant demands.

The chief heat-producing constituents in any of the foregoing fuels are carbon and hydrogen or their compounds.

42. Coal.—Coal is the transformed remains of the rank and luxuriant vegetation of prehistoric ages, and is widely distributed in various localities of the world. The constituents of coal are carbon, hydrogen, oxygen, nitrogen, sulphur, ash, and moisture. Coal varies greatly in composition and characteristics which are dependent upon the changes that the coal has undergone, and the impurities that have entered into it during the transformation.

The heat-producing constituents of a coal are the carbon, the hydrogen and the sulphur. These items collectively are the burnable or combustible part of the fuel. The term *combustible* is usually applied to the part of the fuel that is represented after the moisture and ash have been deducted. Combustible as thus defined includes carbon, hydrogen, oxygen, nitrogen, and sulphur.

43. Analyses of Coal.—Coal analysis data may be obtained by the use of either the *proximate* or the *ultimate* method. Either method is accurate if properly carried out.

The *proximate analysis* is made by mechanical means and separates the coal into four constituent parts, which are by percentages, moisture, volatile matter, fixed carbon, and ash. The sum of these four percentages is equal to 100. A separate determination of the sulphur is usually made and reported along with, but independent of, the other determinations of the proximate analysis.

Moisture.—Coal contains moisture in two forms: that which is inherent in the coal and that which is due to exposure to the weather and other sources of water. All coal used in analyses and calorific determinations should be in equilibrium with the air, so that moisture is neither given to nor received from the surrounding atmosphere. Coal in such a state is said to be air dried. The loss in moisture upon air drying is secured by drying a representative sample of properly

sized coal, at a temperature of 90 deg. fahr., until it shows a loss of weight not to exceed 0.10 per cent per hour.

An air-dried sample is then crushed and finely ground; and further dried in an oven at a temperature of 220 deg. fahr. for a period of one hour. The loss of weight from this sample is then noted. From the data secured for the losses in weight, upon air drying and oven drying, the total moisture percentage of the coal is calculated. It is essential that coal used for oven drying be air-dried before it is finely crushed and ground. If the coal is not air-dried, considerable moisture will be lost during the crushing and grinding operation; thus an appreciable error will result in the determination of the total moisture content.

Example.—A 5.0-lb. sample of coal was air-dried until its weight was constant at 4.68 lb. A finely ground one-gram sample of the air-dried coal was held in an oven at 220 deg. fahr. for one hour. The weight of the one-gram sample after oven drying was 0.976 gram. Find the total moisture content of the coal.

Solution.—

Percentage of moisture lost upon air drying $= \dfrac{(5.0 - 4.68)100}{5} = 6.4$ per cent

Equivalent weight of the original coal represented by 1 gram of air-dried coal $= \dfrac{1 \times 100}{(100 - 6.4)} = \dfrac{1}{0.936} = 1.068$ gram

Total moisture of the coal $= \left(\dfrac{1.068 - 0.976}{1.068}\right)100 = 8.61$ per cent.

Volatile Matter.—Coals contain hydrogen, hydrogen-carbon compounds (hydrocarbons), and other gas-forming ingredients. The combination of these materials, other than moisture, which is driven off by the application of heat to a coal sample held in a covered crucible is known as the volatile matter.

The usual procedure, in the determination of the percentage of volatile matter is to place one gram of a finely powdered air-dried sample of coal in a small platinum crucible fitted with a tight cover. This tightly covered crucible is held in a gas or electrically heated muffle furnace at a temperature of 1700 to 1775 deg. fahr. for exactly seven minutes. The crucible is then removed from the furnace and allowed to cool within a desiccator without disturbing the crucible cover. After the crucible is cold it is weighed and the loss of weight determined. When proper corrections for the moisture content of the coal have been made, the percentage of volatile matter is computed from the data secured.

Ash.—The residue of the completely burned combustible, together with the earthy matter contained in the coal, is designated as ash. The sample used in the volatile-matter determination is burned at a high temperature until it shows no further loss of weight. Care must be taken, in the burning process, that none of the lighter residue is lost from the crucible. The net weight of the residue represents the ash and earthy matter of the sample. This percentage should be adjusted to the original moisture content of the coal.

Fixed Carbon.—The uncombined carbon of the coal is designated as fixed carbon. The percentage is determined by subtracting from 100 the sum of the percentages of the moisture, volatile matter, and ash. Not all of the carbon of the coal is represented by this percentage of fixed carbon as the volatile matter contains a part of the total carbon in combination with hydrogen. Furthermore, not all of the sulphur may have disappeared with the volatile matter.

Sulphur.—Sulphur is an undesirable element in a fuel because the sulphur compounds formed by oxidation may, under certain conditions, produce corrosive mixtures. Furthermore, in many cases sulphur is the cause of the fusing and clinkering of the ash. The sulphur determination is made separately by using a fused sample of the coal. The fused coal is chemically treated and the resulting precipitate collected. The precipitate formed is burned and the loss of weight noted, and from this the sulphur percentage is computed.

44. Ultimate Analysis.—When all of the percentages of the various coal constituents are reported, an ultimate analysis is necessary. In such an analysis the percentages of carbon, hydrogen, nitrogen, and oxygen are determined by the use of chemical apparatus and calculations. The percentages of sulphur, moisture, and ash are secured as in the case of the proximate analysis. Ultimate analyses can be made satisfactorily only by those well trained in the work. When careful studies of combustion are necessary, ultimate analyses of the fuel are essential.

45. Reports of Fuel Analyses.—The analyses of fuels and the heat values of fuels may be reported in three ways:

(1) As received or as fired.

(2) Dry or moisture free.

(3) Combustible, or ash and moisture free.

Coals can best be compared with each other when placed on a dry basis, as their moisture content is variable with the weather and other conditions. The analysis of the wet coal is desirable as it represents the " as-fired condition " of the coal entering the furnace.

Wet coals may be reduced to a dry basis by dividing each of the constituent percentages, except the moisture percentage, by 1 − moisture

(one minus the moisture content expressed decimally). Likewise, the percentages representing the combustible may be obtained by dividing the wet coal values, except moisture and ash, by 1 − moisture and ash (one minus the moisture and ash expressed decimally).

Example.—The proximate analysis of a coal as received gave moisture, 5.02; volatile matter, 35.67; fixed carbon, 52.26; and ash, 7.05 per cent. Express the analysis on a dry basis.

Solution.—

		Per Cent
Volatile matter	= 35.67 ÷ (1 − 0.0502)	= 37.55
Fixed carbon	= 52.26 ÷ (1 − 0.0502)	= 55.03
Ash	= 7.05 ÷ (1 − 0.0502)	= 7.42
Total		100.00

When conversion from the dry basis or combustible basis is to be made to the as-received condition, the dry or combustible values are multiplied by 1 − moisture, or by 1 − ash and moisture; and the percentages of moisture, or moisture and ash added to the respective analyses.

46. Coal Classifications.—A scheme of coal classification is difficult to make up, covering every condition encountered and sharply separating all coals into distinctive classes. Schemes of classification that are in use are based on:

(1) The carbon content.
(2) The ratio of the carbon to the hydrogen content.
(3) The ratio between the volatile matter and the fixed carbon.
(4) The percentage of volatile matter and the vein moisture content.
(5) The unit heat value of the pure coal (termed unit coal) and the unit volatile matter percentage.[1]
(6) The rank of the coal.

The United States Geological Survey uses the term "rank" (of item 6) to designate the present stage of transition in the coal formation. The term rank signifies the completeness of the transition from the first stage, which is peat, to the final stage of almost pure graphite. The word grade is also applied in order that the quality of the rank may be defined with respect to ash, moisture, and sulphur content.

If the Geological Survey classification is used, the progressive stages in the formation of coal are: peat, lignite, sub-bituminous, bituminous, semi-bituminous, semi-anthracite, anthracite, and almost pure graphite.

A classification of coals, based upon the percentages of volatile matter

[1] "The Classification of Coal," by Samuel W. Parr. University of Illinois Engineering Experiment Station Bulletin No. 180, 1928.

and the percentages of moisture of the coal in the vein, is given by Table IV. This classification is a modification of a classification made by Frazer in 1877 in a study of Pennsylvania coals.

TABLE IV

Classification of Coals by Volatile Matter and Vein Moisture

Coal	Volatile Matter, Per Cent	Vein Moisture, Per Cent
Anthracites	Below 5	
Semi-anthracites	5–10	
Semi-bituminous (Pocahontas)	15–32	
Bituminous (Eastern)	25–35	2–4
Bituminous (Mid-Continent)	35–45	6–17
Black lignites sub-bituminous	35–45	17–20
Brown lignites	25–45	20–25

Peats and graphitic coals are not considered, as they are of no consequence as fuels for power purposes. Peat, when freshly dug, contains a large amount of water and cannot be burned until it has been dried. Graphitic coals can be burned only with great difficulty. In the intermediate states between peat and graphite, coal finds a wide application as a fuel. The greater part of the fuel used for power purposes falls under the classifications of the various grades of bituminous coals.

47. Physical Properties of Coals.—The physical characteristics of coals are listed in the following paragraphs.

Anthracite is a very hard coal having a shiny black luster. Anthracite coal is non-coking, and contains a high percentage of fixed carbon with a very small amount of volatile matter. This coal is slow to ignite unless the furnace temperature is high; and it requires a strong draft. Anthracite coal burns without flames or with a very short bluish flame, and is desirable where smokeless combustion is essential. The supply is very limited; thus the cost is becoming high. Fine sizes are available for use in stoker-fired installations. The principal and best field of anthracite coal in the United States is in eastern Pennsylvania.

Semi-anthracite coal occurs in small quantities in various localities of the United States, and is not used commercially as a steam coal. The principal differences from a true anthracite are: less fixed carbon, more volatile matter, less luster, and it burns with longer and more luminous flames.

Semi-bituminous coal is the highest grade of bituminous coal and burns with a very small amount of smoke. This coal is softer than the anthracites, contains more volatile matter, and has a tendency to break up into small sizes during storage or transportation. The latter item is of small consequence in the generation of steam so long as the size is uniform. The principal fields of this coal in the United States are in West Virginia.

Bituminous coals are "soft"; have high percentages of volatile matter; burn with long yellow and smoky flames; and vary greatly in the percentages of volatile matter, moisture, ash, and sulphur contents. The chief distinction between the Eastern bituminous coals and those of the central portion of the United States is in their percentages of volatile matter and moisture content when mined. Both high- and low-ash coals are included. A high-ash coal is usually designated as one having more than 10 per cent ash and earthy matter content; also such coals in many cases have a high-sulphur content. Bituminous coals are classified as free-burning, and caking or coking varieties. Free-burning coals do not swell up and fuse together when ignited, as do the coking or caking coals.

Lignite coals are of two grades, the black and the brown. Black lignites are sometimes classified as sub-bituminous coals, and lack the woody structural appearances of the brown lignites. Black lignites are a very low grade of bituminous coal; they disintegrate rapidly when exposed to the air and require careful attention during storage. The use of lignite fuels is confined largely to the region in which they are produced.

Brown lignites are those coals which are in the transition state between peat and the sub-bituminous grade, and have a woody or often a clay-like appearance; together with low-heat values and high-moisture and high-ash contents. The initial moisture content may be as high as 30 to 45 per cent, but moisture is readily lost upon exposure to the air which may produce rapid disintegration of the fuel. Unless carefully stored, lignites are subject to spontaneous combustion. Fuels of this class are of value for local use only, but give promise of greater usefulness in a pulverized form.

48. Coal Sizes.—Coal taken from the mine during mining operations ranges in size from fine dust to large lumps. Generally the fine sizes contain more impurities and dirt, due to segregation, consequently they do not usually have as high a heating value per pound as the larger sizes. Also, special equipment may be necessary in order to burn them. Therefore, the smaller sizes are lower in price and find a large use for the production of steam in power plants. The preparation and sizing of coal

forms an essential part of the industry of coal production. The sizing operations are carried on by crushing and screening the coal to produce the various market sizes.

The sizes recommended by the American Society of Mechanical Engineers for the various trade names of anthracite coal are given in Table V.

TABLE V

ANTHRACITE COAL SIZES

Trade Name	Diameter of Screen Openings, Inches	
	Through	Over
Broken	$4\frac{1}{2}$	$3\frac{1}{4}$
Egg	$3\frac{1}{4}$	$2\frac{5}{16}$
Stove	$2\frac{5}{16}$	$1\frac{5}{8}$
Chestnut	$1\frac{5}{8}$	$\frac{7}{8}$
Pea	$\frac{7}{8}$	$\frac{9}{16}$
No. 1 buckwheat	$\frac{9}{16}$	$\frac{5}{16}$
No. 2 buckwheat	$\frac{5}{16}$	$\frac{3}{16}$
No. 3 buckwheat	$\frac{3}{16}$	$\frac{3}{32}$
Culm or waste	$\frac{3}{32}$	

In the case of bituminous coals there is no general classification of sizes that is recognized in all localities. The sizing of bituminous coals recommended by the American Society of Mechanical Engineers is placed in two groups. One is for eastern and the other for western coals. The recommended sizes for the various trade names of bituminous coal sizes are given in Table VI.

49. Chemical Properties of Coals.—Coals vary greatly in the percentages of their various constituents, and consequently in the heat that they will produce per pound. This variation is marked not only in the same coal field but also in the same vein and even in the same mine. Analyses of samples of the various ranks of coal, selected and adjusted from United States Bureau of Mines Bulletin 22, are given in Table VII. Reference should be made to the bulletin for a more complete listing of the various coals of the United States.

50. Coal Impurities.—Moisture, ash, oxygen, sulphur, and nitrogen are all coal impurities. The effects of these impurities, either individually or collectively, have a marked influence upon the combustion of the various coals. In general, it is desirable to have a fuel which has a low content of each and all the impurities mentioned.

TABLE VI

BITUMINOUS COAL SIZES

Name	EASTERN. Diameters of Screen Openings, Inches		WESTERN. Diameters of Screen Openings, Inches	
	Through	Over	Through	Over
Run of mine	Unscreened coal as taken from the mine		Unscreened coal as taken from the mine	
Lump	.....	$1\frac{1}{4}$		
Lump, 6 in., 3 in., $1\frac{1}{4}$ in.	.....	.....	.....	6, 3 or $1\frac{1}{4}$
Lump, 6 in $\times$ 3 in	.....	.....	6	3
Lump, 3 in. $\times$ $1\frac{1}{4}$ in.	.....	.....	3	$1\frac{1}{4}$
Nut	$1\frac{1}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$
Slack	$\frac{3}{4}$			
Screenings	.....	.....	$1\frac{1}{4}$	

Nitrogen occurs in limited quantities and is chemically inert. The effect of nitrogen is small, as it acts only as a diluent of the products of combustion.

Sulphur.—Unless sulphur occurs in a free state in the coal it does not have its full heat value. In the sulphate form sulphur is worthless. Sulphur in large quantities is prone to facilitate the fusion of the ash and earthy matter, thus producing a troublesome clinker which is hard to remove from the furnace.

Ash and earthy matter represent waste materials which reduce the heat value of the fuel and which must be removed both from the fires and from the plant at a considerable expense. Ash tends to clog the air spaces in the fuel bed, reducing the rate at which fuel may be burned. Also, certain coals have an ash which fuses at low temperatures and forms clinkers. The clinkering qualities of a coal are very important from an operating standpoint. Clinkers cause furnace-lining deterioration, labor and expense of removal, as well as loss in capacity due to the fusing material sealing over the air openings in the grate.

Moisture represents a part of the coal which has no fuel value. A loss of heat energy occurs in the moisture evaporation, and its escape with the products of combustion.

Oxygen has the general effect of lowering the heat value of a fuel by an amount equivalent to that occasioned by the same percentage of ash.

TABLE VII

Analyses of Coals of the United States

(Abstracted from U. S. Bureau of Mines Bulletin 22)

Rank	State	County	Sample	Proximate Analysis, Coal As Received				Ultimate Analysis, Coal As Received					Heat Value by Calorimeter, B.t.u. per Lb.	
				Moisture	Volatile Matter	Fixed Carbon	Ash	Sulphur	Carbon	Hydrogen	Oxygen	Nitrogen	As Rec'd	Dry
Anthracite	Colorado	Gunnison	Mine	2.70	3.32	88.15	5.83	0.80	85.38	2.98	1.19	1.12	14,099	14,490
Anthracite	Pennsylvania	Lackawanna	Car culm	5.41	7.02	71.79	15.78	0.74	72.65	2.50	2.15	0.77	12,047	12,737
Anthracite	Pennsylvania	Schuylkill	Mine	2.80	1.16	88.21	7.83	0.89	84.36	1.58	1.91	0.63	13,298	13,682
Semi-anthracite	Colorado	Gunnison	Mine	4.86	6.96	81.87	6.31	0.81	79.66	3.43	3.00	1.93	13,468	14,155
Semi-anthracite	Pennsylvania	Sullivan	Mine	3.38	8.47	76.65	11.50	0.63	78.43	3.20	1.86	1.00	13,156	13,617
Semi-bituminous	Maryland	Alleghany	Car run of mine	3.42	17.65	71.84	7.09	0.84	80.17	4.34	2.69	1.45	14,162	14,665
Semi-bituminous	Pennsylvania	Cambria	Car run of mine	3.51	16.82	73.04	6.63	0.94	80.70	4.17	2.79	1.26	14,279	14,800
Semi-bituminous	West Virginia	Fayette	Car run of mine	2.52	21.28	68.20	8.00	0.68	78.71	4.45	4.33	1.31	13,997	14,359
Semi-bituminous	West Virginia	McDowell	Composite mine	3.50	15.0	78.30	3.20	0.60	84.39	4.20	2.73	1.38	14,800	15,330
Bituminous	Alabama	Bibb	Mine	3.16	31.05	59.56	6.23	1.20	78.28	4.98	4.78	1.37	14,141	14,602
Bituminous	Colorado	Boulder	Car run of mine	18.68	34.88	40.45	5.99	0.55	57.46	3.99	12.18	1.15	10,143	12,472
Bituminous	Illinois	Franklin	Mine	11.50	26.70	52.67	9.13	0.60	65.40	4.01	8.28	1.08	11,410	12,893
Bituminous	Illinois	La Salle	6 in. lump car	12.39	36.89	41.80	8.92	3.92	61.29	4.47	8.01	1.00	11,399	13,012
Bituminous	Illinois	Sangamon	Car run of mine	16.00	32.41	37.82	13.77	4.05	53.89	3.77	7.63	0.91	9,940	11,833
Bituminous	Illinois	Williamson	Mine	7.88	31.20	49.89	11.03	2.99	66.50	4.26	6.30	1.04	11,741	12,746
Bituminous	Indiana	Green	Car run of mine	10.30	36.31	41.64	11.75	4.23	61.00	4.24	7.42	1.06	11,218	12,508
Bituminous	Iowa	Marion	Car run of mine	14.21	33.17	37.40	15.22	4.66	54.08	3.93	6.59	1.31	10,019	11,678
Bituminous	Kansas	Crawford	Car run of mine	4.99	32.68	49.36	12.97	4.28	67.34	4.43	4.91	1.08	12,242	12,884
Bituminous	Kentucky	Hopkins	Car run of mine	7.92	36.09	45.93	10.06	3.52	65.29	4.51	7.30	1.40	12,022	13,055
Bituminous	Ohio	Hocking	Mine	9.72	32.44	53.41	4.43	0.54	69.50	4.62	9.94	1.25	12,247	13,567
Bituminous	Oklahoma	Pittsburg	Mine	2.09	27.59	50.25	20.07	5.73	63.66	4.23	2.89	1.33	11,695	11,945
Bituminous	Pennsylvania	Fayette	Car run of mine	5.13	27.87	58.29	8.71	0.86	73.13	4.34	6.33	1.50	13,365	14,089
Bituminous	Pennsylvania	Westmorel'nd	Car run of mine	3.39	31.79	56.46	8.36	1.05	74.42	4.69	6.70	1.39	13,699	14,178
Bituminous	Tennessee	Campbell	Car run of mine	5.38	34.54	53.03	7.05	0.99	72.41	4.90	7.55	1.72	13,048	13,788
Bituminous	Virginia	Lee	Car run of mine	4.06	34.93	56.28	4.73	1.20	76.59	4.87	7.31	1.24	13,826	14,411
Bituminous	West Virginia	Kanawha	Car run of mine	2.82	32.20	56.95	8.03	1.38	76.74	4.85	4.84	1.34	13,766	14,166
Bituminous	West Virginia	Mingo	Car run of mine	2.86	33.23	58.08	5.83	0.67	78.38	4.69	6.14	1.43	14,105	14,521
Lignite	Colorado	Boulder	Mine	19.15	30.82	44.27	5.76	0.25	56.38	3.80	13.58	1.08	9,616	11,893
Lignite	North Dakota	McLean	Lump	35.96	31.92	24.37	7.75	1.15	41.43	2.54	9.96	1.21	7,069	11,038
Lignite	North Dakota	Williams	Car run of mine	38.92	25.54	30.15	5.39	0.48	39.34	2.56	12.63	0.68	6,739	11,034
Lignite	Texas	Wood	Car run of mine	33.71	29.25	29.76	7.28	0.53	42.52	3.04	12.13	0.79	7,348	11,084
Lignite	Wyoming	Sheridan	Mine	29.33	28.58	36.42	5.67	1.21	45.09	3.01	14.80	0.89	7,783	11,012
Lignite	Washington	King	Mine	17.51	31.17	38.55	12.77	0.37	52.11	3.90	12.26	1.08	9,331	11,312

51. Coal Storage.—Large users of coal are compelled to provide facilities for storing quantities of coal for use in periods of heavy consumption or market shortages. Certain difficulties are encountered in coal storage, which are the effects of weathering upon the heat value, disintegration of the coal, and the likelihood of fire due to spontaneous combustion.

The suitability of a coal for storage purposes is influenced largely by the impurities contained within the coal, together with the size of the coal. Coals may be safely stored by the proper selection of the coal, by storage under water, or by piling the coal properly. Coal stored in piles should be tightly compacted and the fines mixed in with coarser material; there should not be regions of coarse material adjacent to regions of fine material; air pockets and air circulation within the pile, as well as all sources of heat should be avoided. Coal piles should not be made too deep and the temperatures of the interior of the pile should be watched closely, as excessive or dangerous temperatures will necessitate re-piling. Coals high in oxygen and sulphur are harder to store satisfactorily.

52. Pulverized Coal.—Coal when finely crushed and ground to a powder of dust fineness will float in the air and may be burned somewhat like a gaseous fuel. The limits of fineness are that 95 per cent of the powdered coal will pass through a 50-mesh screen, and 65 per cent through a 200-mesh screen.

Two methods of preparing and burning coal in such a condition are in use. One system involves crushing, drying, grinding, and storing the coal in airtight bins from which the coal is fed into the furnace with a strong blast of air. The other scheme is known as the unit system in which each furnace is provided with a separate pulverizer. Coal of suitable moisture content is fed to the pulverizing mill and then blown directly into the furnace. The unit system permits the use of higher moisture contents in the coal as it passes through the powdering mill. The equipment of both systems is described in Chapter VIII, Boiler Auxiliaries.

The advantages of the use of powdered coal are these: a low grade of fuel may be used; combustion is more complete as the air supply is under better control; coal and ash-handling labor costs are reduced; and flexibility of operation, under varying load conditions, is possible.

The disadvantages of operation are the initial cost of installation; difficulty of crushing the coal to a uniform size; danger of clogging the feed pipes and storage bins, due to moisture when the unit system is not used; high maintenance cost of the furnace linings; slagging of furnace walls and tubes; ash accumulations on the tubes; and the problem of the

ash removal from the chimney gases. A pulverized fuel plant may prove objectionable in a city due to the escape of ash with the chimney gases.

53. Wood.—Wood as a fuel is not of general commercial importance and its use is confined to more or less isolated conditions. The moisture content of wood is variable, being as high as 50 per cent when freshly cut and from 10 to 20 per cent after having been exposed to the air for some time. Woods are classified as *hard* and *soft*, the heat values ranging from 8300 to 9000 B.t.u. per pound of combustible. The desirability of a wood, as a fuel, is influenced by its moisture content and its density or weight per cubic foot. As woods are usually sold by volume instead of weight, the heat value per unit of measure varies with the dryness and the density of the air-dried wood.

54. Oils.—The principal liquid fuels are crude oils or derivatives from crude petroleum. These range in heat value from 18,500 to 20,000 B.t.u. per lb. The principal oil fields of the United States are located in Pennsylvania, Ohio, Indiana, Illinois, Kansas, Oklahoma, Texas, and California. Oils are classified according to the base left after distillation, which may be paraffin, asphalt, or olefine. The latter group is not found in this country, while the asphalt-base oils are found chiefly in Texas and California.

Improved gasoline-refining methods have reduced the number of heavy distillates from the paraffin-base oils; so that their price is often prohibitive for power purposes. The asphalt-base crudes provide most of the fuel oils for power purposes.

Crude oils and their distillates are hydrogen-carbon compounds, together with moisture, oxygen, sulphur, nitrogen, and silt. The moisture content of crude oil, direct from the well, may range from 1 to 30 per cent; whereas the distillates have a much lower moisture content. The carbon content ranges from 80 to 85 per cent and hydrogen from 12 to 15 per cent.

In comparison with coal, oil as a fuel has the following advantages: (1) cleanliness, (2) flexibility of operation, (3) elimination of ash handling, (4) reduction of labor in firing, (5) ease of control, (6) high-operating efficiency, (7) reduced maintenance charges on boilers, but not on furnaces.

The following disadvantages are to be considered in the use of oil: (1) initial cost of the storage tanks, burners, heaters and pumps, furnace upkeep greater, (2) adequacy of the supply, (3) the cost of the supply.

55. Gaseous Fuels.—These fuels may be classified as those of natural origin and those of artificial origin. They are ideal fuels, but in most cases are too valuable for the generation of steam for power.

Gas is a very desirable fuel for internal-combustion engines from

which a higher thermal efficiency may be obtained than in the case of steam-using equipment.

Fuels such as blast-furnace and by-product coke-oven gases are available in iron and steel plants. When there is no available market for such gases, they may be burned under steam boilers or used in internal-combustion engines.

Natural gas occurs in various localities and usually consists of a high percentage of methane (a carbon-hydrogen compound), hydrogen, carbon monoxide, and small percentages of nitrogen, oxygen, etc. The heat value of natural gas ranges from 850 to 1100 B.t.u. per cubic foot.

Blast-furnace gases are low in heating values, which range from 80 to 100 B.t.u. per cubic foot. The heat values are due largely to the carbon monoxide (CO) content which is about 25 to 35 per cent. Other constituents are hydrogen, carbon dioxide (CO_2), nitrogen, and a very small percentage of oxygen.

By-product coke-oven gases range from 400 to 500 B.t.u. per cu. ft. in heat value, and are much more valuable than blast-furnace gases. They contain considerable percentages of hydrogen and methane, together with low percentages of carbon dioxide (CO_2), nitrogen, oxygen, and carbon monoxide.

Producer and illuminating gases are suitable for use in internal-combustion engines, but are not of commercial importance in the generation of steam. Producer gas is formed by the combustion of any grade of coal, coke, or peat with a deficient air supply. Incomplete combustion results with the formation of a considerable quantity of carbon monoxide. Other constituents of the gas are hydrogen, hydrogen-carbon compounds, nitrogen, together with small percentages of oxygen, and carbon dioxide.

Illuminating or water gas is usually made by the decomposition of steam into hydrogen and carbon monoxide, in the presence of incandescent carbon from coal or coke. The gas is enriched by hydrocarbons from crude oils or petroleum products.

56. Calorific Value of Fuels.—The heat liberated by the complete and rapid oxidation (or burning) of a unit weight or volume of a fuel is the *heating* or *calorific value* of the fuel. In the case of solid and liquid fuels the heating value is expressed in B.t.u. per lb. In the case of gaseous fuels the heating values are expressed in B.t.u. per cu. ft. of gas measured at a temperature of 60 deg. fahr. and a pressure of 30 in. of mercury.

Fuels which contain hydrogen or hydrogen-carbon compounds have two heating values, the *higher* heating value, and the *lower* heating value. The combustion of hydrogen produces superheated water-vapor, and in the ordinary furnace this superheated vapor escapes at the tempera-

ture of the chimney gases. Consequently, the superheat and the latent heat of vaporization of the vapor formed are not available for heating purposes. The *lower-heating value* is the net heat liberated per pound of fuel after the heat necessary to vaporize and superheat the steam formed from the hydrogen, as well as that from the fuel, has been deducted. The *higher heating value* of a fuel is the one indicated by any sort of a fuel calorimeter. In the case of calorific determinations, the steam produced in the calorimeter is condensed and the heat of superheat and the latent heat of vaporization are recovered.

The heating value of a fuel may be found by two methods: by calculation, and by the use of a fuel calorimeter. When the ultimate analysis of the fuel is known the heating value may be computed. The most satisfactory method of determining the heating value is to burn a quantity of the fuel in a calorimeter and to measure the temperature rise induced in a given quantity of water. When the proper corrections are applied for the calorimeter parts, radiation, etc., the heat value per unit of the fuel may be determined from the calorimeter data.

57. Heating Value of a Fuel by Calculation.—In the calculation of the heating value of a fuel from the ultimate analysis only carbon, hydrogen, and sulphur are considered as heat producers. Not all of the hydrogen, indicated by the ultimate analysis, is available for the production of heat. Two atoms of hydrogen unite with one of oxygen to form water.

$$2H + O = H_2O \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

In the molecular form, using the atomic weights of hydrogen and oxygen, this is expressed as

$$2H_2 + O_2 = 2H_2O \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

$$(2 \times 2 \times 1) + (16 \times 2) = 36$$

$$1 \quad + \quad 8 \quad = 9$$

A part of the hydrogen of the fuel is rendered unavailable for heating by the oxygen which is in combination with it in the form of inherent moisture. The hydrogen available for the production of heat is

$$\left(H - \frac{O}{8}\right)$$

H and O represent the proportionate parts of hydrogen and oxygen, by weight, per pound of the fuel. The heats liberated by the complete combustion of one pound of each of the following elements are: carbon, 14,600 B.t.u.; hydrogen, 62,000 B.t.u.; and pure sulphur, 4050 B.t.u.

Professor S. W. Parr recommends the use of a value 5000 B.t.u. per lb. for the form of sulphur appearing in coals.[1]

Dulong's formula for the heating value of a fuel based on the proportions by weight of the total carbon, the available hydrogen, and the sulphur is expressed

$$F = 14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4050\,S \quad . \quad . \quad . \quad (67)$$

Only for coals of some regions are the heat values calculated by the use of Dulong's equation in close agreement with those obtained by the use of a calorimeter. Certain coals, such as Illinois coals, may contain calcium carbonate, which may lead to an erroneous ash determination so that the percentages of oxygen and hydrogen are also incorrect. A discrepancy between the calculated and the calorimeter-determined heating value results.

58. Solid- and Liquid-Fuel Calorimeters.—Fuel calorimeters are of two general types: those which employ the oxygen bomb, and those which utilize sodium peroxide as an agent to supply the oxygen for combustion within the closed cartridge of the calorimeter. The latter type of calorimeter gives results that are satisfactory for many purposes, but the oxygen bomb type is recognized as the standard calorimeter in the determination of the heating value of fuels. Standard calorimeters are the Mahler, Atwater, Emerson, and the Parr Illium-alloy oxygen bomb calorimeter. It is essential that any oxygen bomb calorimeter should have a non-corrosive bomb or a bomb with a perfect non-corrosive lining.

During the combustion of the fuel within the calorimeter, nitric and sulphuric acids are formed. These acids result from the nitrogen of the fuel, the nitrogen of the air, and the sulphur of the fuel. Unless the bomb lining is non-corrosive the acids attack the metal, and heat is liberated which will be credited to the fuel as it is impossible to make corrections for such variable reactions. Bomb linings of porcelain, spun gold, or spun platinum are satisfactory so long as they entirely cover the inside of the bomb. Platinum makes an ideal bomb material but the price is prohibitive. A platinum substitute, Illium (a nickel, chromium, copper alloy), developed by Professor S. W. Parr, is a satisfactory bomb material.

The cross section of a Parr oxygen bomb made of Illium, of the Mahler-Atwater type, is shown by Fig. 19. The essential parts of the bomb are the cover A, the cover clamp nut C, the supports DD, the fuel tray G, the rubber gasket B, and the check valve E. The supports

[1] "The Classification of Coal," by S. W. Parr, University of Illinois Engineering Experiment Station Bulletin No. 180, 1928.

DD serve as electrodes and a fine fuse wire is looped between them as shown so that it dips into the fuel held in the fuel tray *G*. The bomb is capable of holding oxygen under a pressure of 350 to 450 lb. per sq. in.

The assembled bomb, containing the fuel and oxygen under 20 to 30 atmospheres pressure, is placed in the calorimeter as shown in Fig. 20. Part *H* is a metal water container of oval cross section. The bomb *F* is placed slightly to one side of the center of the oval can so that the motor-driven water stirrer *S* may have space. The thermometer *T*, for the measurement of the temperature rise of the water surrounding the bomb, is placed at the opposite side of the oval can. The body of the calorimeter, including the cover and bottom, is made of indurated fiber with an air space in the walls. This body provides considerable protection against a flow of heat to the calorimeter water from the surrounding air or a loss of heat from the water to the surrounding air. Part *I* represents one of the two clamps used for holding the cover in place. The electrical circuit through the fuse wire is completed through the binding posts *JJ*. An external source of low voltage current is applied. The shell of the bomb *F* makes electrical contact at *K*, and a flexible electrical connection is attached to one electrode *D* at *J*.

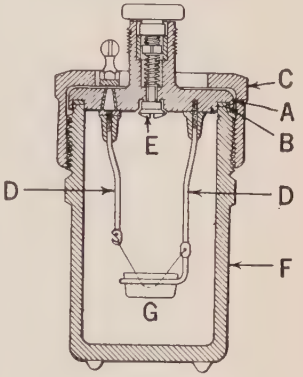


Fig. 19.—Oxygen bomb details.

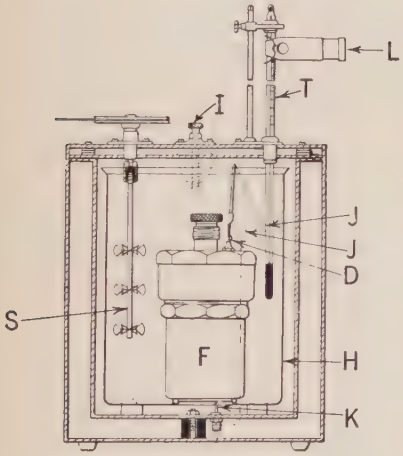


Fig. 20.—Oxygen bomb calorimeter assembly.

A heating-value determination is made with the calorimeter in the following manner. A carefully weighed amount, approximately one gram, of finely pulverized air-dried coal is placed in the fuel tray *G*. A loop of No. 34 B.S. gage pure-iron wire is attached to the electrodes so that this fuse wire just touches the fuel. After the bomb cover is in place the bomb is carefully and slowly charged with oxygen.

The bomb is then placed in the oval container located within the calorimeter body and 2000 grams of distilled water are added to the can. The water-stirring mechanism is set in motion and preliminary temperature observations taken with the calorimeter thermometer. The fuel is ignited by closing the electrical circuit for a very short period of time. Observations of the water temperature are recorded until a period of five minutes after the end of the combustion period has elapsed. The preliminary and final temperature observations are used to determine the radiation corrections.

The total temperature rise of the calorimeter water, corrected for radiation effects, is multiplied by the weight of water including the water equivalent of the metal parts of the calorimeter bomb and can. The water weight and the water equivalent of the calorimeter are best expressed in pounds. If the thermometer readings were in degrees fahrenheit, the product of the equivalent water weight and the temperature rise represents the total heat in B.t.u. liberated within the calorimeter. Additional corrections are necessary to compensate for the heat produced by the formation of nitric and sulphuric acids, and also the combustion of the fuse wire. From these derived data the heat value per pound of the fuel may be computed.

The procedure in the case of calorific determinations of liquid fuels is the same as for solid fuels. However, in the case of volatile liquid fuels it is necessary to weigh the fuel in closed glass bulbs, which are broken in the calorimeter fuel tray just before the bomb is closed. The loss of the volatile constituents of the fuel is thus prevented during the weighing operation.

59. Gas Calorimetry.—A much-used type of gas calorimeter is the Junkers calorimeter, shown by Fig. 21. The gas, whose heat value is to be determined, is passed through a calibrated meter D and its pressure and temperature noted. A pressure regulator in the meter inlet line maintains the pressure of the gas nearly uniform as it flows to a carefully adjusted Bunsen burner B located within the calorimeter body. The calorimeter body C is a vertical cylindrical chamber having a water jacket through which water is allowed to flow under a constant head. Calibrated thermometers TT are placed in the inlet and outlet water connections. The heat produced by the gas is determined by accurately weighing the water passing through the calorimeter while a given quantity of gas is burned. Frequent readings of the inlet and outlet water temperatures are taken during this period and the average temperature rise of the water is determined. The product of the temperature rise and the weight of water passing through the calorimeter, divided by the

cubic feet of gas burned gives the heating value of the fuel per cubic foot as measured. This result may be corrected to heating value per cubic foot at the standard conditions of 60 deg. fahr. temperature and 30 in. of mercury pressure. The heating value thus determined is the higher value. If the lower heating value is desired, it is necessary to collect the moisture condensed from the products of combustion and

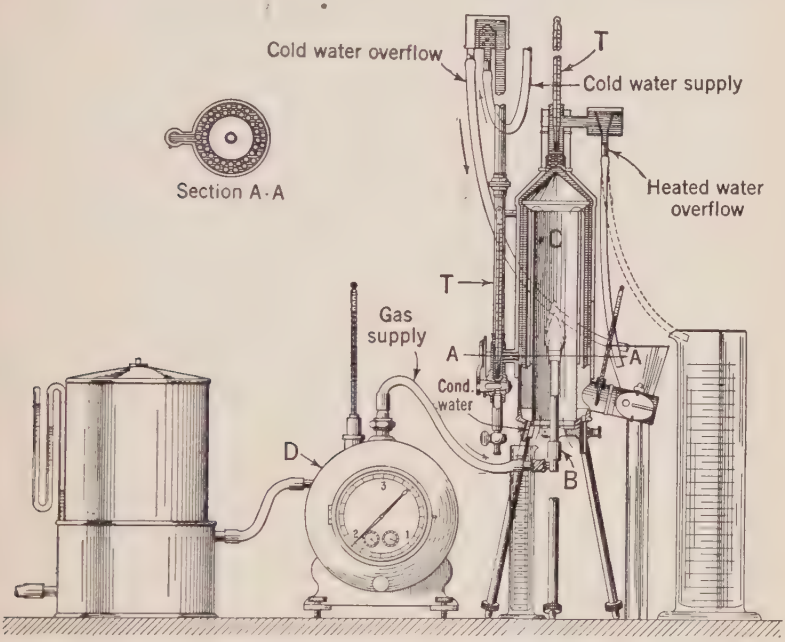


FIG. 21.—Junkers gas calorimeter.

make a correction for the heat liberated by the condensation of the vapor formed during combustion.

PROBLEMS

1. A 5-lb. sample of coal as received weighed 4.83 lb. after air drying. One gram of the air-dried sample after oven drying had its weight reduced to 0.964 gram. Find the total moisture percentage of the coal.

2. A 4-lb. sample of coal as received weighed 3.58 lb. after air drying. The weight of one gram of air-dried coal after oven drying was 0.940 gram. Find the total moisture of the coal in per cent.

3. Express the proximate analysis of a Schuylkill County, Pennsylvania, anthracite coal on a dry basis. Also on a moisture- and ash-free basis.

4. Express the ultimate analysis of a Franklin County, Illinois, bituminous coal on a dry basis. Also on a moisture- and ash-free basis.

5. A bituminous coal has a total moisture content of 15.71 per cent. The proximate analysis percentages on a dry basis are: volatile matter, 37.30; fixed carbon, 42.90; and ash, 19.80. Express the analysis on an as received basis.

6. The heat value of the coal in problem 5 is 11,312 B.t.u. per pound of dry coal by calorimeter determination. What is the heating value per pound as received? Also per pound of combustible?

7. Calculate by Dulong's equation the heating value of one pound of Fayette County, West Virginia, semi-bituminous coal.

8. Calculate by Dulong's equation the heating value of one pound of Hocking County, Ohio, bituminous coal.

CHAPTER V

COMBUSTION AND FLUE GAS ANALYSIS

60. The Combustion Process.—The three principal heat-producing elements of fuels are carbon, hydrogen, and sulphur, separately or in combinations. The reaction that releases heat energy from the elements is one of oxidation, in which the necessary oxygen is supplied from the air. Unless the process is a rapid one, the amount of heat energy released in a given period of time is small. Therefore, in the combustion of fuels for power purposes the oxidation processes must be rapid and must produce a considerable quantity of heat. Where the fuel exists in a dust, gaseous, or atomized form, under certain conditions, exceedingly rapid oxidation or combustion may produce an explosion. In general, the combustion reactions take place when the burnable parts of the fuel have been reduced to a gas and unite with the oxygen of the air for further oxidation. Except in specific cases this process is slow enough to prevent an explosive action.

61. Combustion Prerequisites.—Three conditions are necessary to produce and maintain the complete combustion process of fuels. The first requirement is an adequate supply of oxygen procured from a sufficient but not excessive air supply. An inadequate quantity of air provides a deficient amount of oxygen, while an excessive amount of air produces a large volume of flue gases with a resultant loss of heat to the chimney. Second, the mixture of the oxygen and the burnable parts of the fuel must be thorough. Third, the temperature maintained must be high enough to gasify the fuel, to ignite the mixtures, and to maintain the process.

Combustion prerequisites may also be stated in another manner, as air, time, temperature, and turbulence. Time is required to reduce the fuel components to a gaseous state and completely oxidize them. A sufficiently high temperature must be maintained to gasify the fuel and ignite the products formed. Turbulence of the gases formed is essential for intimate mixing of the volatile products with the oxygen of the air.

In the case of gaseous fuels, the fuel-air mixture can be more easily controlled and, unless the furnace and burner are of exceedingly poor design, the ignition temperatures can be maintained without difficulty.

Atomization or vaporization of liquid fuels is the first essential in their combustion, together with sufficient air and the proper ignition temperature. Unless the atomization is thorough the combustion will be incomplete and an inefficient process results.

Solid fuels when pulverized may be made to float in the air and may be burned in a manner similar to gaseous fuels. Solid fuels when in lumps or pieces of various sizes undergo somewhat different states during the process of complete combustion. The stages are: the ignition or kindling temperature at which the volatile constituents begin to distill off, the expulsion and burning of the volatile materials, and the complete oxidation of the remaining burnable parts, leaving the ash and refuse.

62. Gaseous Products of Combustion.—Pure carbon when burned with an adequate and thorough mixture of air will liberate 14,600 B.t.u. per pound of carbon. The resulting gas formed is carbon dioxide, CO_2. If the air supply is deficient or if a thorough mixture of the carbon and the air is not obtained, the gas formed will be carbon monoxide, CO. When the latter condition exists only 4440 B.t.u. are liberated per pound of carbon and the process is an inefficient one in furnace operation, due to the loss of a valuable unburned portion of the fuel.

Pure hydrogen burns to water vapor. Hydrogen alone or in combination with carbon in hydrogen-carbon compounds (hydrocarbons), may escape if the combustion process is incomplete. Hydrocarbons completely burned disappear as water vapor and carbon dioxide. Sulphur usually burns to sulphur dioxide, SO_2, in the ordinary furnace.

Moisture in the coal is converted into water-vapor or superheated steam and usually represents a heat loss. In the case of fuels of very fine sizes, and where the draft is heavy, moisture may be an aid in securing higher operating efficiencies, as unburned fuel particles may be prevented from being discharged at the stack. Moisture may also cause certain coals to cake and thus reduce the amount of fine unburned coal appearing in the ash and refuse, thus producing a resultant saving. The idea is prevalent that adding water to coal will increase the heat output of a given amount of the fuel, due to the decomposition of the water into hydrogen and oxygen while in the furnace. The thought is that the hydrogen thus released will provide additional heat when burned, but this is not true as the decomposition of the water requires as much heat as can be recovered by the combustion of the hydrogen. Furthermore, the hydrogen burns back to water vapor which escapes as superheated steam. This represents a distinct heat loss in the flue gases.

63. Combustion Stages.—Combustion proceeds as follows in the case of a solid fuel such as coal being burned upon a grate. Immediately after a fresh charge is thrown upon the incandescent fuel bed the

moisture disappears as steam. The volatile matter, which includes hydrogen and hydrogen-carbon compounds, next distills off and if their combustion is complete, water vapor and carbon dioxide are formed. These products escape, together with sulphur dioxide, and the oxygen and nitrogen are released from the fuel. If the combustion of the volatile matter is incomplete, unburned hydrogen-carbon compounds may escape as well as water vapor, carbon monoxide, and carbon dioxide. The remnants of the fuel left, after the moisture and the volatile matter have disappeared, are the fixed carbon and the ash.

The fixed carbon in the lower part of the fuel bed, where the air supply is sufficient, immediately burns to carbon dioxide. However, as this CO_2 passes upward through the fuel bed, the oxygen supply is generally deficient in proximity to the large quantities of carbon. As a consequence, a reducing instead of an oxidizing action takes place and the carbon dioxide is changed to carbon monoxide. This carbon monoxide, as it passes from the fuel bed, is again oxidized to carbon dioxide; provided the air supply is sufficient and a thorough air-gas mixture is secured at a temperature sufficient for ignition. Sometimes it is necessary to admit air above the fire in order to secure efficient combustion.

An inadequate supply of air, a small furnace volume, and cold surfaces or low temperatures above the fire hinder further combustion and promote the formation of soot, smoke and carbon monoxide, as well as the escape of unburned hydrogen and hydrocarbons.

64. Furnace Requirements.—The design of the furnace plays an important part in the completeness of combustion secured. The various fuels require different types of furnace construction and different boiler settings. Unless the furnace is properly designed a thorough mixture of air and the gases from the fuel, and a sufficiently high temperature for their combustion cannot be maintained.

Adequate combustion space should be provided in the furnace, or in a chamber adjacent to it, to allow an intimate mixture of the gaseous products with the requisite amount of air. In modern large boiler installations the boilers are set high above the level at which the fuel is admitted to the furnace, and a large combustion space is provided within the furnace itself. In older boiler settings a refractory lined combustion chamber is provided behind the bridge wall where the combustion process is completed. In such an arrangement the air and gases are diffused as they pass over the bridge wall. In the latter setting both the furnace and the combustion chamber should be separated from the boiler-heating surfaces by refractory arches. Ordinarily the reactions of combustion must be completed before the gases leave the combustion chamber and strike the comparatively cool heating surfaces of the

boiler. Secondary combustion, in and around the tubes of a boiler, is unsatisfactory and likely to be variable in its completeness.

An exception, to the rule of not having comparatively cool surfaces for the unburned gases to play upon, is the case of the large modern stoker-fired or powdered coal-fired boiler furnaces. These furnaces invariably are large in volume and the boilers are driven at several times their normal rating. In such furnaces very high combustion temperatures are secured; so that the refractory linings of the furnaces are likely to be melted, or badly eroded by molten slag and ash from the coal. In such cases water pipes may be embedded in the furnace walls just behind the inner face. Water from the boiler is circulated through these pipes which serve as additional boiler-heating surface. These water pipes are for the purpose of keeping the furnace linings below their melting point. Another method used in large modern boiler-furnaces is that of the water wall which forms the side enclosure of the furnace. Water from the boiler is circulated through this additional heating surface. These water tubes have given good results even though they bring a comparatively cool surface in proximity to the burning fuel. The radiant type of steam superheater is also used for the same purpose. However, these devices are adapted only to large furnaces using preheated combustion air; moreover the whole combustion process must be carefully handled.

Another prerequisite of a good furnace is the control of the air admitted to the furnace. A deficiency of air, or a great excess of air, supplied to the furnace results in loss of efficiency. The furnace must be tight; so that air cannot leak into it at points other than where air is required. Air leakage may cause the furnace temperature to be reduced below the ignition temperature of the gaseous products, or else it may cause an excessive amount of heat to be carried away by the greatly diluted flue gases. When the flue gases are diluted with excess air, their temperature is lowered and the heat transmitted to the water of the boiler is less.

From the standpoint of efficient and thorough combustion, as well as rapid transfer of heat from the flue gases through the boiler tubes and plates to the water within the boiler, high furnace temperatures are desirable. On the other hand high furnace temperatures may result in rapid deterioration of the furnace linings, with the resultant maintenance charges, and the unavailability of the boiler for use during furnace lining repairs.

Stoker-fired boiler furnaces of medium or large size should have a volume of $\frac{1}{3}$ to $\frac{1}{2}$ cu. ft. for each pound of fuel burned per hour. A

fair value for the volume of pulverized fuel furnaces is 1 cu. ft. of internal volume for each pound of powdered coal burned per hour.

65. Combustion Chemistry.—A modern conception of the atom is that it is made up of minutely small bodies called electrons and protons. The electrons carry a negative charge of electricity and the protons a positive charge. The atom consists of a nucleus about which the electrons revolve. The nucleus is composed of both electrons and protons with a greater number of the latter. The relation of the electrons and protons is extremely difficult to change in the atom of any element. Therefore, the atom is taken as the smallest division of an element that will enter into a combination to form a compound.

The atoms of each element have a definite and fixed mass or weight. The hydrogen atom is the lightest and its weight has been taken as unity or 1. The atomic weights of the chemical elements usually involved in combustion reactions are given in Table VIII. The numerical figures given are not the exact atomic weights of the elements but are sufficiently accurate for all combustion calculations.

TABLE VIII

NAMES, SYMBOLS, AND ATOMIC WEIGHTS OF ELEMENTS INVOLVED IN COMBUSTION REACTIONS

Name	Symbol	Atomic Weight	Name	Symbol	Atomic Weight
Hydrogen	H	1	Oxygen	O	16
Carbon	C	12	Sulphur	S	32
Nitrogen	N	14			

Any symbol of Table VIII when taken singly as H, the hydrogen symbol, represents one atom. If the symbol were written as H_2; then two atoms of hydrogen are involved. When atoms enter into a combination the smallest particle, that can be formed by the combination, is spoken of as the molecule. The molecules of hydrogen, oxygen, and nitrogen, are signified by the symbols H_2, O_2, and N_2. In these cases two atoms are necessary to form a molecule of the element. If the number two is prefixed to any of the above symbols, as $2H_2$, it indicates that two molecules are involved, in this case two hydrogen molecules.

So long as the atoms of an element remain uncombined with the atoms of other elements, their physical and chemical properties are unchanged. When the atoms of an element enter into chemical com-

bination with other atoms, the physical and chemical properties of the molecule formed are different from those of the molecules of the elements entering into the combination. The properties of the molecule formed are distinctive for the particular combination. The combinations of atoms to form molecules are dependent upon the conditions existent at the time of the combination. Hence, the molecules formed may be predetermined by the proper control of the conditions attendant to their formation from atoms.

Avagadro's Law deduced experimentally shows that, *when at the same temperature and pressure, equal volumes of all gases contain the same number of molecules.*

A convenient unit of measurement is the pound-mol or *mol*, which is a weight in pounds equal to the molecular weight of the element or compound. A mol of all elements, or compounds, contains the same number of molecules. One mol of all gases occupies the same volume under standard conditions of 32 deg. fahr. and 29.92 in Hg abs. pressure. The volume of one mol of gas is the product of its molecular weight and the volume per pound of the gas. Thus the volume of one mol of oxygen, O_2, at standard conditions is $11.21 \times (16 \times 2) = 358.7$ cu. ft.

When a reaction is written in the molecular form, the coefficients represent the relative number of cubic feet of the substance entering into the reaction as well as the weights. In a mol-volume V of a mixture of three gases, whose fractions of the total volume are v_1, v_2 and v_3, there are v_1 mol volumes of gas *1*, v_2 mol-volumes of gas *2*, and v_3 mol-volumes of gas *3*, when all exist at the same temperature and pressure. The total weight of the mol is mV, when m is the molecular weight of the gas mixture. If the molecular weights of the gas constituents are represented by m_1, m_2, and m_3, the weights of the various gases of the mixture are m_1v_1, m_2v_2 and m_3v_3. The total weight of the gas mixture is $mV = m_1v_1 + m_2v_2 + m_3v_3$, and the proportional parts of the mixture w_1, w_2 and w_3, by weight, are,

$$w_1 = \frac{m_1v_1}{mV},\ w_2 = \frac{m_2v_2}{mV} \text{ and } w_3 = \frac{m_3v_3}{mV}.$$

66. Formation of Carbon Dioxide and Carbon Monoxide.—Carbon when oxidized may burn to either carbon dioxide, CO_2, or carbon monoxide, CO, depending upon the amount of oxygen entering into the reaction. In the case of the formation of CO the combustion of the carbon is incomplete. The formation of carbon dioxide is represented by the equation

$$C + O_2 = CO_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

in which one combining weight of carbon unites with two combining weights of oxygen to form carbon dioxide. If the atomic weights are substituted, in equation 68, the molecular weight of the carbon dioxide is found to be

$$12 + (2 \times 16) = 44.$$

If the numerical equation is divided by 12, the atomic weight of carbon, the relative proportions of the components are

$$1 + 2\tfrac{2}{3} = 3\tfrac{2}{3},$$

which signifies that $2\frac{2}{3}$ lb. of oxygen are required for the complete combustion of each pound of pure carbon.

Air by weight contains 23.15 per cent oxygen and 76.85 per cent nitrogen and other inert gases. As air is the usual source of oxygen for combustion processes, the weight of air required for the complete combustion of one pound of carbon is $(2\frac{2}{3} \div 0.2315) = 11.52$ lb.

When carbon does not take up the maximum amount of oxygen in combustion and CO is formed, the relation of the carbon to the oxygen is

$$2C + O_2 = 2CO \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

In this case two atoms of carbon unite with two atoms of oxygen to form two molecules of carbon monoxide CO. The weight of 2CO is

$$(2 \times 12) + (2 \times 16) = 56$$

and the relative proportions of the constituents are

$$1 + 1\tfrac{1}{3} = 2\tfrac{1}{3}$$

Thus in the formation of CO only $1\frac{1}{3}$ lb. of oxygen are required per pound of carbon, and the weight of air necessary is

$$(1\tfrac{1}{3} \div 0.2315) = 5.76 \text{ lb.}$$

Carbon monoxide may be burned to CO_2 under the favorable conditions of an additional oxygen supply and a temperature sufficient to facilitate the process. In this case two molecules of the monoxide unite with one molecule of oxygen to form carbon dioxide.

$$2CO + O_2 = 2CO_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

The process requires $(2\frac{2}{3} - 1\frac{1}{3}) = 1\frac{1}{3}$ lb. of additional oxygen per pound of carbon.

67. Hydrogen and Sulphur Combustion.—In the chapter on Fuels, it was shown that the burning of hydrogen to water vapor proceeds according to the following equations:

$$2H_2 + O_2 = 2H_2O \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

$$4 + 32 = 36$$

$$1 + 8 = 9$$

Thus the complete burning of one pound of hydrogen requires 8 lb. of oxygen, and 9 lb. of water vapor are formed. The air required is $(8 \div 0.2315) = 34.56$ lb. per pound of hydrogen consumed.

Sulphur burns in air to sulphur dioxide, SO_2, and the process is expressed

$$S + O_2 = SO_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (72)$$

$$32 + 32 = 64$$

$$1 + 1 = 2$$

Each pound of sulphur burned requires one pound of oxygen, so that the air requirements are $(1 \div 0.2315) = 4.32$ lb. per pound of sulphur completely burned.

68. Theoretical Amount of Air for Combustion.—The theoretical amount of air for combustion may be computed from the ultimate analysis of a fuel. The parts of a solid or a liquid fuel available for heat production are the fixed carbon, the available hydrogen, and the sulphur. When the fractional parts, per pound of fuel, of each of the heat-producing constituents are multiplied by the air necessary for the complete combustion of a pound of that constituent, the sum of the air requirements is the theoretical weight of air necessary to burn one pound of the fuel.

$$W_{ta} = 11.52C + 34.56\left(H - \frac{O}{8}\right) + 4.32S. \quad . \quad . \quad . \quad (73)$$

69. Composition of Flue Gases.—Gaseous combustion products, which pass from the furnace or boiler setting to the chimney, are known as flue gases. The fuel burned, the conditions of combustion, and the tightness of the furnace and boiler setting affect the flue gas composition.

Flue gases produced by the burning of a solid fuel, such as coal, usually contain percentages of carbon dioxide, carbon monoxide, oxygen, nitrogen, free carbon, ash, unburned hydrogen-carbon compounds, sulphur dioxide, and superheated steam. If the combustion process has

been absolutely complete, free carbon, carbon monoxide, hydrogen, and hydrocarbons will be absent.

Difficulty is encountered in controlling the combustion process so that it is complete with just the theoretical amount of air. Some excess air is required to insure complete combustion; thus oxygen usually appears in flue gas analyses in varying amounts. Free unburned carbon and ash dust, appearing in a finely divided state in the flue gases, give the gases colors ranging from a light gray to deep black. The term smoke is then applied to the flue gases. The appearance of free carbon in flue gases is due to the fact that, even though ample air has been provided, the gases have been below the ignition temperature before complete oxidation of all of the burnable parts has been completed. This is especially true of coals or fuels containing oily or tarry compounds. Separately, the free carbon particles of smoke do not represent a great heat loss, but frequently the completeness of the combustion process is indicated by the smoke color. Dark to black smoke is indicative of a carbon monoxide and an unburned hydrogen-carbon content. The two latter items represent a greater heat loss in the chimney gases than soot or free carbon does.

70. Flue Gas Analyses.—Flue gas analyses are based on the percentages of the various constituents by volume in the dry gases. Usually the analyses include only the percentages of carbon dioxide, oxygen, carbon monoxide and nitrogen. More complete analyses include the percentages of hydrogen and hydrocarbons that may appear in the gases.

Dry air is composed of 20.99 per cent of oxygen and 79.01 per cent nitrogen and other inert gases by volume. The complete combustion of one pound of carbon with the exact theoretical amount of air would produce a flue gas whose carbon dioxide content would be 20.99 per cent, and whose nitrogen content would be the same as that of dry air. Oxygen and carbon monoxide would not appear in the flue gases. If the carbon is incompletely oxidized, carbon monoxide will appear in the analysis. If an excess of air is supplied, the gases will have a free oxygen content. In any event when pure carbon is burned, the sum of the carbon dioxide, the oxygen, and the carbon monoxide percentages will equal 20.99 per cent, if the percentage of CO is small.

Practically all fuels contain hydrogen and hydrogen-carbon compounds. The combustion of the hydrogen produces water vapor which does not appear in the dry gases when analyzed. The oxygen for the production of the water vapor is obtained from air which leaves a nitrogen residue, and which increases the percentage of nitrogen in the flue gases. Also, the small amount of nitrogen contained in the fuel has the same

effect. Coals which contain hydrogen and nitrogen, therefore, give flue gases which have nitrogen contents of 81 to 85 per cent. The sum total of the other flue gas constituents in an actual analysis may range from 15 to 19 per cent.

Flue gas analyses should be made of representative gas samples. The sample should be drawn from the furnace or boiler setting through a single open-ended metal or fused quartz tube. During the time that the sample is being drawn, the open end of the tube should be placed in various locations in the region from which a gas sample is desired. Sampling-tube locations at bends and in pockets below dampers are to be avoided. When a sample is drawn from the furnace, or the first passes of a boiler, or any region of extremely high temperature, a fused quartz tube must be used. Metal sampling tubes are likely to be oxidized by the oxygen of the gas sample at high temperatures. The reaction results in a reduction of the oxygen content of the gas sample, and the data obtained will not be representative of the flue gases being discharged.

The gas sample drawn for analysis should be collected in a container, over mercury or water well saturated with the flue gases. The sample should be aspirated into the container by a strong, continuous, and uniform process. Some sort of a filter or separator for moisture, tar, and dirt is desirable in the line between the sampling tube and the gas container. The gas analysis should be made immediately after the sample is collected. In many cases it is preferable to draw the filtered gas directly from the sampling tube into the analyzing apparatus.

71. Flue Gas Analyzers.—Many commercial power plants or boiler rooms are provided with equipment which indicates or records continuously the CO_2 content of the flue gases. Some CO_2 recorders or indicators are delicate and easily deranged and require much attention. The indications given by nearly all recorders lag more or less behind the flue gas production. All CO_2 recorders should be checked frequently for accuracy. The CO_2 content alone is not an indication of efficient combustion, as high percentages of CO_2 may also be accompanied by appreciable percentages of CO, due to the small percentage of excess air being supplied. Therefore, more complete analyses are essential at frequent intervals.

A portable analyzer which may be used for test work and for checking CO_2 recorders and indicators is either the Orsat apparatus or some of its modifications. A typical Orsat apparatus is shown by Fig. 22. The essential parts of the Orsat are the measuring burette A, the leveling bottle F, the gas cleaner H, and the absorption pipettes B, C, D, and E. The measuring burette A, the filter H, and the absorption pipettes are

interconnected by means of a manifold having accurately ground cocks *I*. The measuring burette, *A*, is water-jacketed to prevent temperature and density changes of the gas while the analysis is in progress. The filter *H* may be filled with spun glass or similar material.

Pipette *B* is partially filled with a 40 per cent solution of caustic potash, KOH. This reagent absorbs CO_2 and is the first one used during an analysis. Pipette *C* contains an alkaline solution of pyrogallic acid. The pyrogallol absorbs oxygen and is made up of a mixture of equal volumes of 33 per cent KOH and 25 per cent pyrogallic acid solutions. The two pipettes *D* and *E* contain an acid solution of cuprous chloride which is used for the absorption of carbon monoxide, CO. Pipettes *D* and *E* should contain some fine metallic copper to keep the solution energized.

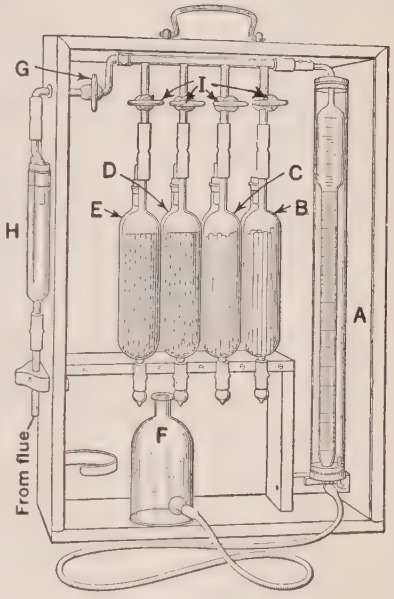


FIG. 22.—Orsat apparatus.

Each absorption pipette contains a number of small glass tubes, which provide a greater amount of wet surface, to facilitate the gas absorption during the time that the flue gas is held in the pipette. The flue gas during analysis is successively passed through pipettes *B*, *C*, and *D* or *E*. The analysis must be made in this order, and the absorption of each constituent must be complete before the gas is exposed to the next reagent. The reagents in the various pipettes should be fresh and should be protected from the air when not in use. All connections between the various parts of the apparatus must be gas tight. Frequent tests for leakage from the apparatus are essential.

A gas analysis is made by the following procedure. With the reagents drawn to the proper marks on the capillary tubes and the cocks of pipettes *B*, *C*, *D* and *E* closed, cock *G* is opened to the air and the leveling bottle *F* elevated to drive air or gases out of the apparatus. Cock *G* is then opened to the flue gas supply and a sample is drawn into the apparatus by lowering the leveling bottle *F*. Gas should be drawn in and expelled two or three times to clear the Orsat of air and vitiated gas. When the final sample of a little more than 100 c.c. is drawn, the leveling bottle must be brought up slowly and carefully with cock *G* open to the

air; so that the water level in the bottle and the burette coincide at the zero mark. When the water levels have been thus brought into coincidence at the zero mark, the cock G should be closed. Exactly 100 c.c. of flue gas are then entrapped in the Orsat measuring burette at atmospheric pressure. The cock on pipette B is then opened and the gas driven into it by raising the leveling bottle F. The action of the CO_2 absorption may be somewhat hastened by lowering and raising the leveling bottle. After some minutes the gas should be drawn back into burette A and the reagent in B brought to its initial mark on the capillary tube. The cock on pipette B is then closed and the water levels in F and A are brought into coincidence. The decrease in volume of the gas represents the percentage of CO_2 absorbed. The absorption of CO_2 in pipette B should be continued until the percentage readings are constant.

The absorption of oxygen and carbon monoxide in pipettes C and D is accomplished in a manner exactly the same as was employed for CO_2. The nitrogen content, N_2, of the flue gas is obtained by subtracting from 100 the sum of the percentages of CO_2, O_2, and CO.

72. Actual Weight of Combustion Air. The analysis of the flue gases formed; the ultimate analysis of the fuel burned; and the molecular weights of the principal constituents of the flue gases provide data for calculating the actual amount of air used per pound of fuel. The percentages of CO_2, O_2, CO, and N_2 of the flue gases are to be taken by volume, while the carbon content in the fuel is by weight.

If 100 mol-volumes of flue gases are considered, the percentages of the flue gas analysis represent the mol-volumes of each constituent present. The weight of nitrogen is $28 \times N_2$, and the weight of carbon appearing as CO_2 and CO is $12(CO_2 + CO)$. Therefore, the weight of nitrogen per pound of carbon is

$$\frac{28N_2}{12(CO_2 + CO)}.$$

As 1 lb. of air contains 0.769 lb. of nitrogen, the air, W_c, per pound of carbon, is

$$W_c = \frac{28\,N_2}{12(CO_2 + CO) \times 0.769} \quad . \quad . \quad . \quad . \quad . \quad (74)$$

If the carbon content C_f of the fuel were completely burned, the actual air per pound of fuel would be

$$W_{cf} = W_c \times C_f$$

or

$$W_{cf} = \frac{28\,N_2}{12(CO_2 + CO) \times 0.769} \times C_f \quad . \quad . \quad . \quad . \quad (75)$$

There is likelihood of unburned carbon appearing in the ash and refuse from solid fuels. Under such conditions, calculations involving the use of flue gas analyses should be based on the carbon actually burned per pound of fuel. The actual carbon consumed per pound is

$$C_1 = \frac{W_f C_f - W_r C_r}{W_f \times 100} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

C_1 = carbon burned per lb. of fuel, lb.
W_f = weight of fuel fired, lb.
C_f = carbon content of the fuel fired, per cent.
W_r = weight of ash and refuse from W_f lb. of fuel, lb.
C_r = carbon content of the ash and refuse, per cent.

Equation 75 may then be written

$$W_{aa} = \left[\frac{28\,N_2}{12(CO_2 + CO) \times 0.769}\right] \times \left[\frac{W_f C_f - W_r C_r}{W_f \times 100}\right] . \quad . \quad (77)$$

W_{aa} = the actual weight of air used per lb. of fuel fired, lb.

If extreme accuracy is required the percentage of nitrogen of the coal should be divided by 76.9, and the weight of air equivalent should be deducted from W_{aa}. This correction represents the air equivalent of the nitrogen content of the flue gases produced by the presence of nitrogen in the fuel.

73. Weight of Excess Air.—Due to furnace and burner limitations, more than the theoretical amount of air is always required for the combustion of a fuel. The difference in amount of air actually used per pound of fuel and that theoretically required is the excess air used. Excess air is usually expressed as a percentage.

$$E_a = \frac{W_{aa} - W_{ta}}{W_{ta}} \times 100 \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

E_a = percentage of excess air.
W_{aa} = actual air used per lb. of fuel, lb.
W_{ta} = theoretical air required per lb. of fuel, lb.

The percentage of excess air required for good combustions may range from 10 to 60. Often in burning solid fuels, holes in the fire, thin fires, strong draft, or leaky boiler and furnace settings may cause the actual percentage of excess air to be large. Improper adjustments of oil or gas burners may allow a great volume of air to pass through the furnace.

Excess air above that absolutely essential to good combustion always represents a distinct and often a large amount of heat lost in the chimney

gases. The extra air dilutes the flue gases and lowers their temperature; so that less heat is transferred to the water in the boiler. Due to the increased weight of flue gases formed, the amount of heat carried away is increased.

74. Weight of Flue Gas Per Pound of Fuel.—Based on reasoning identical with that used to express the actual air used per pound of fuel, the weight of the dry gases formed per pound of carbon is

$$W_g = \frac{44CO_2 + 32O_2 + 28(CO + N_2)}{12(CO_2 + CO)} \quad . \quad . \quad . \quad . \quad (79)$$

The symbols CO_2, O_2, CO, and N_2 all represent percentages by volume as determined by the flue gas analysis.

Equation 79 may be simplified as

$$W_g = \frac{44CO_2 + 32O_2 + 28[(100 - CO_2 - O_2 - N_2) + N_2]}{12(CO_2 + CO)}$$

$$= \frac{11CO_2 + 8O_2 + (700 - 7CO_2 - 7O_2 - 7N_2 + 7N_2)}{3(CO_2 + CO)}$$

$$= \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}$$

The actual dry gas formed per pound of fuel is found by $C_1 \times W_g$.

$$W_{dg} = \left[\frac{W_f C_f - W_r C_r}{W_f \times 100}\right]\left[\frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}\right] \quad . \quad . \quad (80)$$

75. Heat Loss Due to the Formation of Carbon Monoxide.—The formation of carbon monoxide represents a distinct heat loss occasioned by incomplete combustion. This loss per pound of fuel is

$$h_{co} = 10{,}160 \times \frac{12CO}{12(CO_2 + CO)}\left[\frac{W_f C_f - W_r C_r}{W_f \times 100}\right] \quad . \quad . \quad (81)$$

PROBLEMS

1. Calculate the theoretical amount of air required for the combustion of a Franklin County, Illinois, bituminous coal.

2. Calculate the theoretical amount of air necessary for the complete combustion of a Westmoreland County, Pennsylvania, bituminous coal.

3. During a boiler test 14,000 lb. of Franklin County, Illinois, bituminous coal were burned. Ash and refuse collected were 1520 lb. Carbon content of the ash 20 per cent. The average flue gas analysis percentages were CO_2, 12; O_2, 7; CO, 0; and N_2, 81. What was the actual amount of air used per pound of fuel fired?

4. A boiler test gave the following data: time of test, 10 hr.; coal fired, Kanawha, West Virginia; pounds of coal used, 40,000; ash and refuse, 3650 lb.; percentage of carbon in the ash 15; average flue gas analysis — CO_2, 15; O_2, 3.3; CO, 0.9; N_2, 80.8. Find the actual air used per pound of coal.

5. Find the percentage of excess air used in the case of problem 3.

6. Find the percentage of excess air used in the case of problem 4.

7. Find the weight of dry gases formed per pound of fuel fired in the case of problem 3.

8. Find the weight of dry gases formed per pound of fuel fired in the case of problem 4.

9. What is the heat loss due to the formation of carbon monoxide in the case of problem 4?

CHAPTER VI

HEAT POWER PLANTS

76. Power Production.—Mechanical power, as commercially produced, is derived from heat energy (liberated by burning some fuel) or by the action of flowing water. As only consideration of heat power plants falls within the scope of this book, hydraulic or water power plants will not be discussed. In the normal production of power, heat energy contained in the fuel is constantly supplied at one end of the line of plant apparatus. Mechanical or electrical power is delivered at the other end of the system. The transformation of energy is accomplished in either *steam plants* or *internal-combustion engine plants.* Both types of plants require a complete assembly of equipment to produce the desired change.

77. Prime Movers.—In any case, the machine which actually converts heat energy into mechanical energy is known as the *prime mover.*

Steam plants employ as prime movers either turbines or reciprocating engines or both. The steam used is generated in a boiler or boilers by burning the fuel in a furnace or furnaces which are an integral part of the boiler setting. The steam formed by the vaporization of water in the boiler serves as a heat carrier, which is readily conveyed through pipe lines to the prime movers. Other apparatus such as pumps, heaters, condensers, stokers, fans, etc., may be required for the proper operation of the plant. The prime movers of steam plants may be operated either *non-condensing* or *condensing.*

In non-condensing operation the steam exhausted from the prime movers is discharged at atmospheric pressure or at pressures greater than atmospheric. Condensing-plant prime movers exhaust their discharge steam into a condenser in which the pressure is less than atmospheric, and in which the steam is reduced to water. In neither non-condensing nor condensing operation is a prime mover (using steam) able to abstract enough of the heat energy carried by the steam to reduce it completely to water. The principal advantages of condensing operation are the increased amount of energy abstracted per pound of steam, and the greater amount of power that may be developed by a given size of engine or turbine.

Internal-combustion engine plants operate with the combustion of the fuel taking place within the prime mover. The fuels commonly used by internal-combustion engines are: natural gas, producer gas, blast-furnace gas, coke-oven gas, kerosene, gasoline, crude petroleum or its distillates. These plants usually show a better utilization of the heat energy of the fuel, as a number of thermal losses are avoided. When a selection of internal-combustion engines as the prime movers of a plant is considered, attention must be given to the following items: initial cost, maintenance charges, depreciation, and the limited amount of power that may be developed on a given floor space.

Dependent upon the fuel used, internal-combustion engines require a certain amount of auxiliary apparatus for their proper operation. The auxiliary apparatus to be included in the plant may be: fuel pumps, fuel heaters, storage tanks, gas producers, gas coolers, gas scrubbers, tar extractors, and gas holders.

78. Cycle of Operation of a Non-condensing Plant.—A diagrammatic section of a small non-condensing steam power plant is illustrated by Fig. 23. Fuel is charged by hand upon the grate under the boiler where it is burned with a liberation of heat energy. Air, supplying oxygen for the combustion process, flows from beneath the grate up through the fuel bed. The hot gaseous products formed pass to the rear of the boiler setting and then through the boiler tubes or flues to the stack. In this manner the hot gases are brought into contact with a considerable amount of boiler-heating surface. A part of the heat liberated by the combustion of the fuel is transferred through the heating surface to heat and evaporate water supplied to the boiler. Not all of the heat content of the fuel is utilized in the boiler, as there are certain inevitable losses. The losses include: heat carried away by the flue gases in the stack, unburned volatile constituents, radiation, and unburned fuel in the ash and refuse.

The steam produced in the boiler flows from the boiler through piping and a steam separator to the engine. The function of the separator is to remove the greater portion of any water particles that may be carried by the steam. The separated water is returned by a trap to the boiler feed supply in the feedwater heater. The dried steam is expanded in the engine cylinder, where heat energy of the steam is converted into mechanical energy, as the steam acts against the engine piston. The power developed by the engine is transmitted through the engine crankshaft to an electric generator, where the conversion into electrical energy is made. The major portion of the electrical energy of the generator is delivered, as the plant output, at the switchboard.

The expanded steam is exhausted from the engine at a reduced

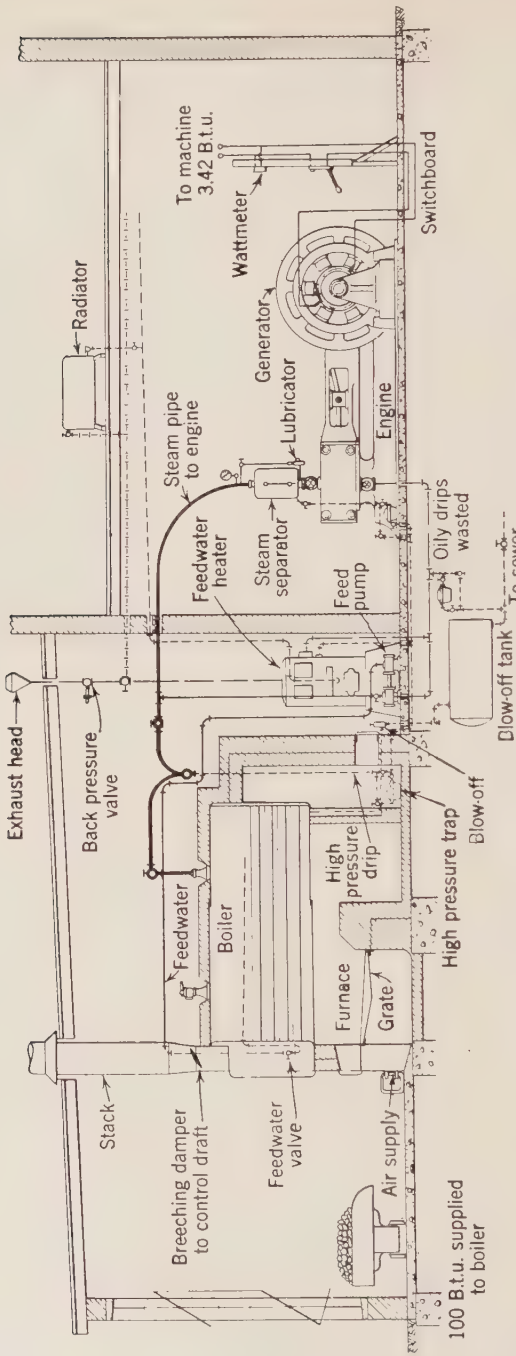


FIG. 23.—Diagrammatic arrangement of a non-condensing steam power plant.

pressure and temperature. The exhaust steam is carried to a feedwater heater, where a portion of it is utilized in heating the water supplied to the boiler. A feed pump is used to force the heated water into the boiler. Excess exhaust steam may flow from the water heater to the radiators of a heating system, or to the atmosphere.

From the coal pile to the switchboard, in each step of the process and in each machine, there are thermal and mechanical losses which reduce the amount of the available energy that is delivered as useful output. In plants similar to that of Fig. 23, the useful output delivered at the switchboard may be 3 to 5 per cent of the energy contained in the fuel when the exhaust steam is discharged to the air. If the excess exhaust steam is used in a heating system, the percentage of the total heat of the fuel utilized in power production and heating may be increased.

Fig. 24 illustrates a modern non-condensing plant in which equipment for handling, storing, and feeding the fuel is installed, as well as ash-conveying equipment.

79. The Condensing Plant Cycle.—The plant equipped with a condenser, Fig. 25, operates in a manner similar to the non-condensing plant, except in the matter of exhaust steam disposal. In condensing operation, the steam discharged from the prime movers is reduced to water in the condensers. This process produces a considerable reduction of the back pressure against which the prime mover works and promotes a better economy in the use of steam and fuel. All, a part, or none of the condensed steam may be returned to the boilers, depending upon the type of condensers used.

Fig. 26 illustrates a modern steam turbine plant in which considerable auxiliary apparatus is installed to conserve and utilize the heat energy of the fuel. The fuel used may be either oil or powdered coal for the plant shown. More detailed descriptions of plant apparatus will be given later.

The overall efficiency, or ratio of the plant heat output as work to the heat input, ranges from 7 to 27 per cent for condensing plants. The latter figure is obtainable only in a very few large modern stations, in which every effort is made to secure very efficient operation.

80. Internal-Combustion Engine Plant.—Plants using internal-combustion engines may burn any of the fuels listed in Article 77, provided that the engine is properly designed to use the fuel. The plant shown in elevation by Fig. 27 employs a Diesel internal-combustion engine to drive an electric generator. Diesel engines burn heavy fuel oils, which are forced into the engine cylinder at the proper time by a fuel pump.

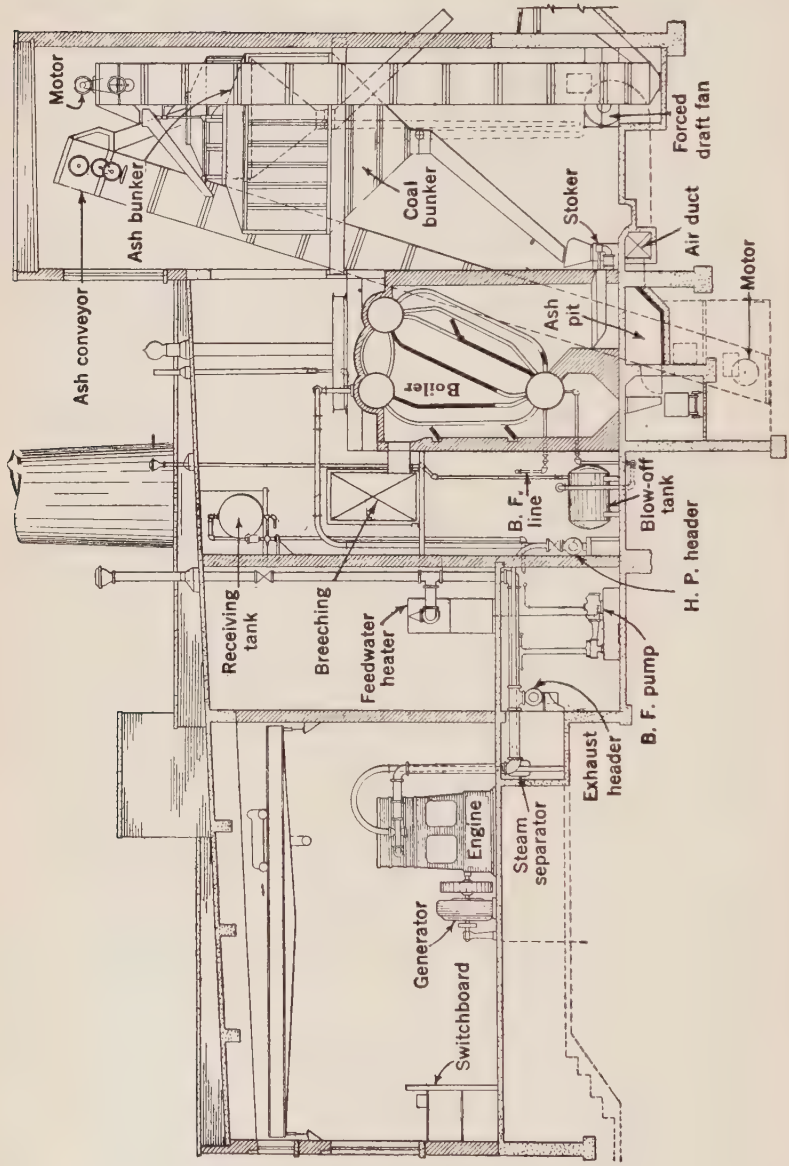


FIG. 24.—Sectional elevation of a non-condensing steam power plant, Marygrove College.

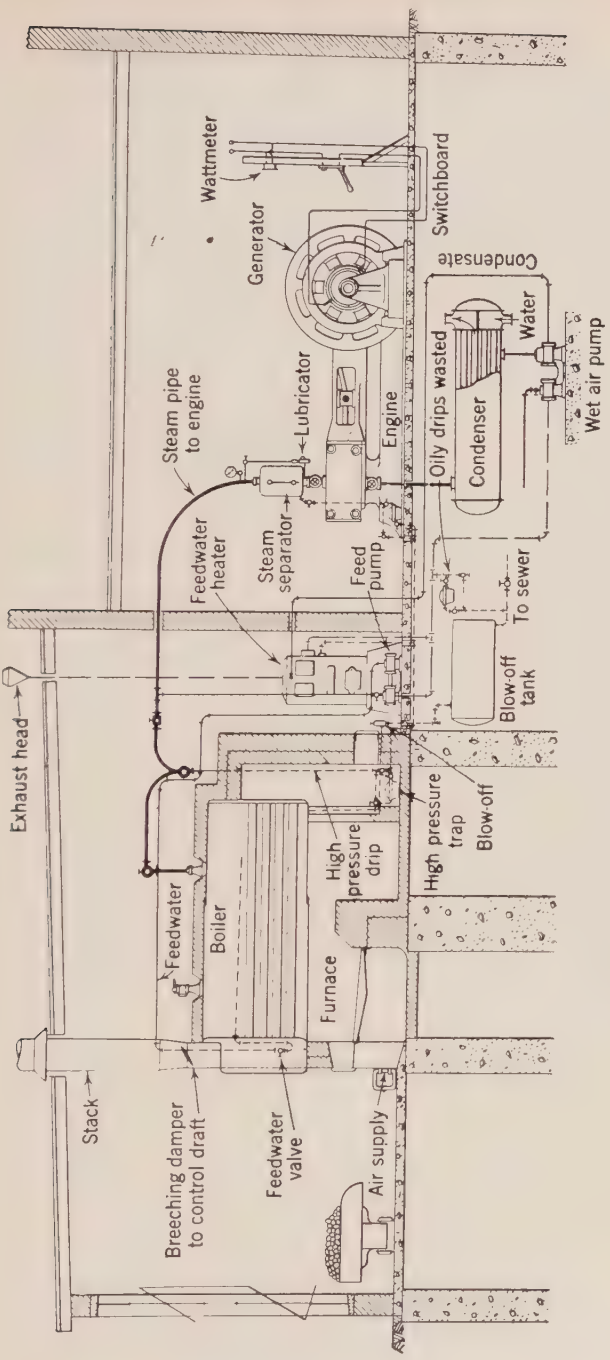


FIG. 25.—Diagrammatic arrangement of a condensing steam power plant.

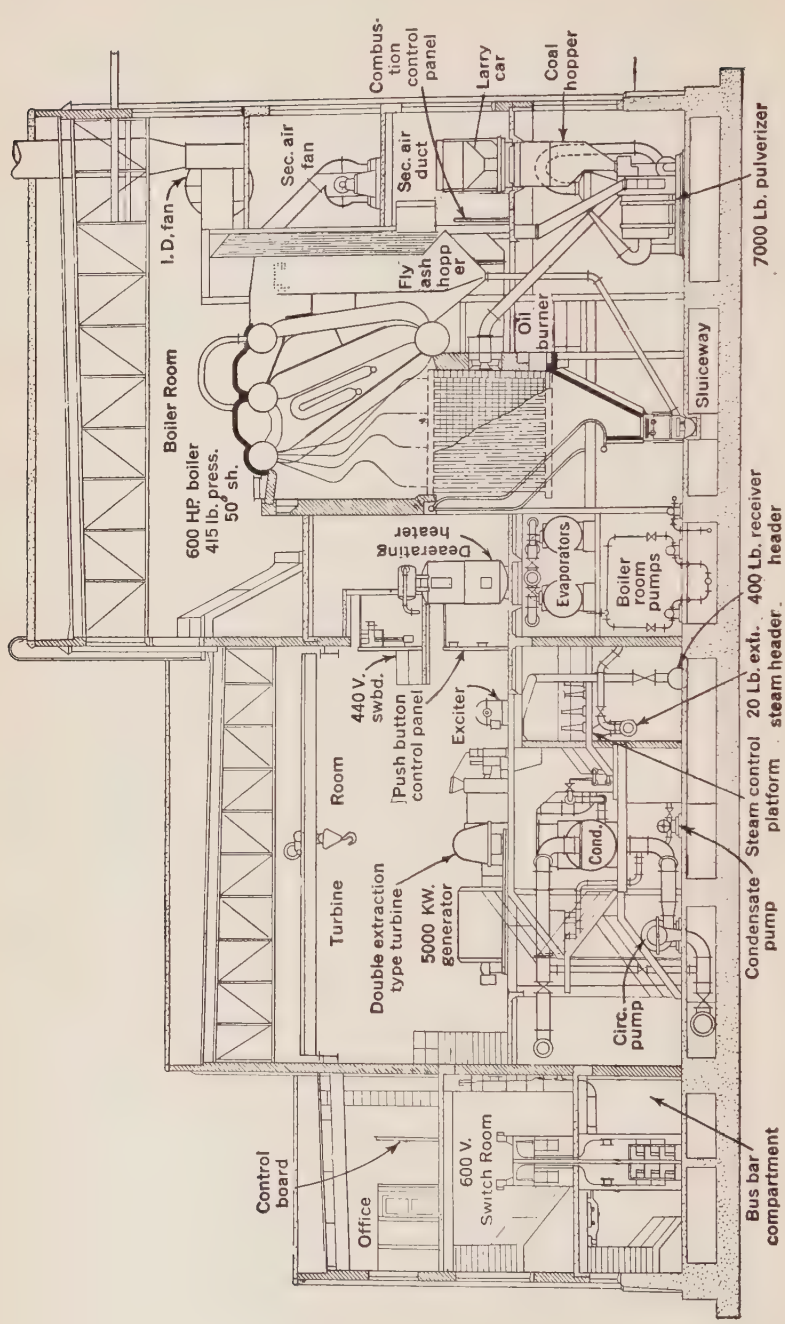


FIG. 26.—Elevation of a condensing steam power plant, Sorg Paper Company.

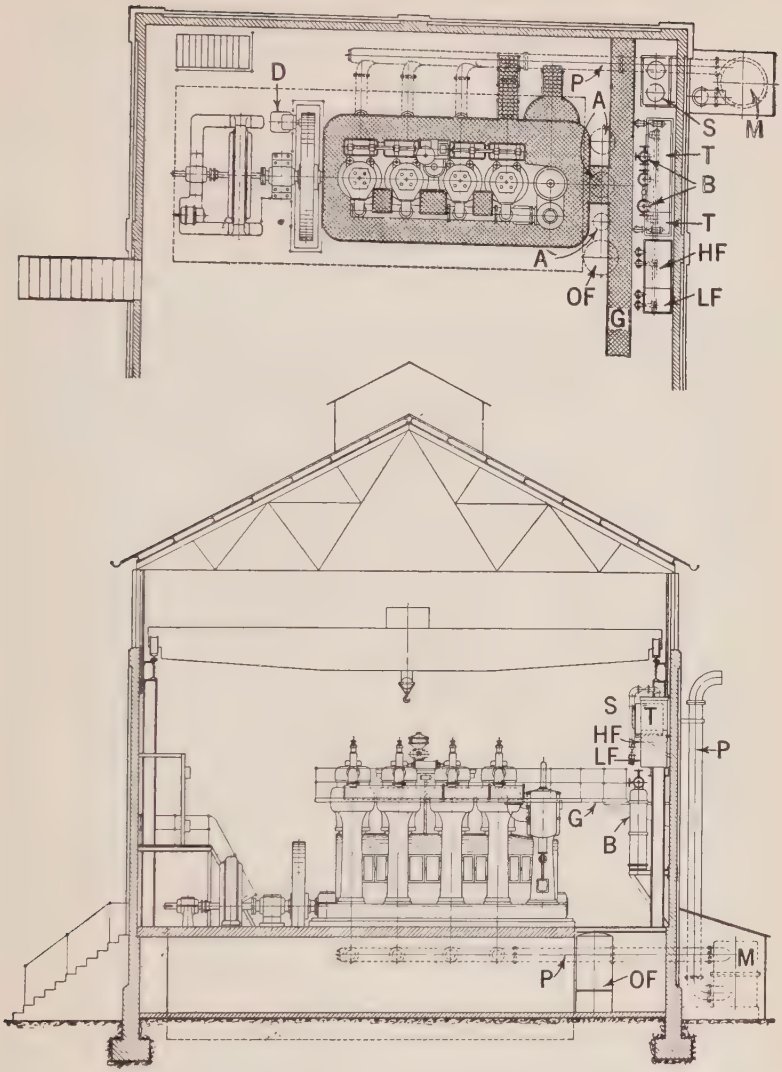


FIG. 27.—Plan and elevation of a Nordberg Diesel engine plant.

The cycle of operation of the engine illustrated is:

(1) Air is drawn into the engine cylinder.
(2) The air is compressed to a high pressure and temperature within the cylinder.
(3) Fuel is injected into the cylinder at the end of the air compression stroke. The fuel is ignited and burned. Expansion

of the gases formed occurs during the working stroke. The expanding gases act against the engine piston and work is done by them.

(4) Exhaust of the burned gases from the engine cylinder to the air.

Internal-combustion engines are built in single or multi-cylinder units, which produce power in one or both ends of each cylinder. The power developed by the engine shown is transmitted to the shaft of an electric generator, where it is converted into electrical power.

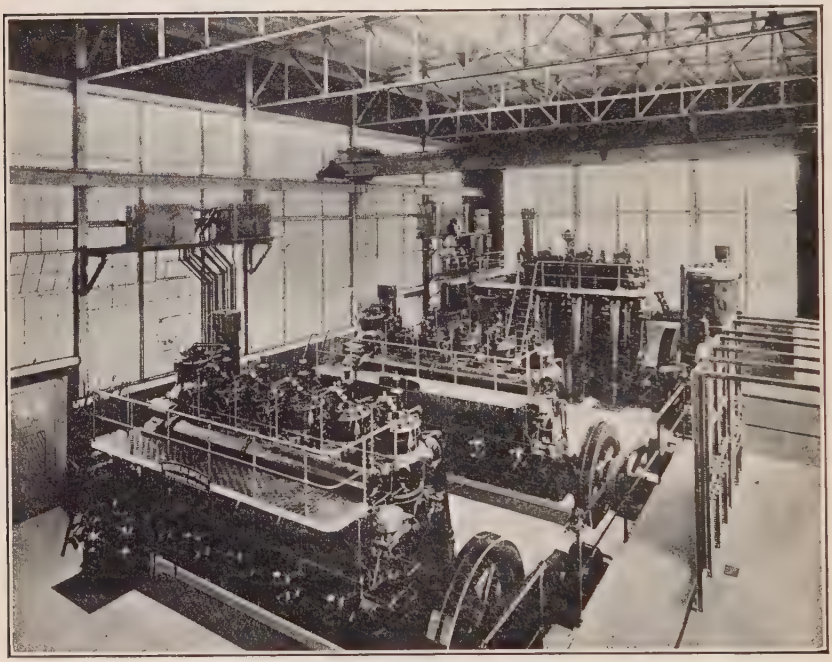


FIG. 28.—Interior of a Diesel engine plant.

The principal parts of the plant, of Fig. 27, are:

Starting device A; bottles for storing compressed air used in engine starting B; barring gear D; lubricating oil pumps and filter tanks OF; fuel filters for both heavy and light oils HF and LF; overhead lubricating oil tanks and coolers S; fuel storage tank T; exhaust piping P; and exhaust muffler M.

An interior view of an internal-combustion engine plant, in which Diesel engines are used to drive an air compressor and electric generators, is shown by Fig. 28.

About 10 to 30 per cent of the heat energy of the fuel is converted into useful output by internal-combustion engine plants.

CHAPTER VII

STEAM BOILERS

81. Development.—One hundred years ago, a steam pressure of 60 lb. per sq. in. was considered dangerously high. In recent years a boiler pressure of 3200 lb. per sq. in. was obtained in an experimental plant at Rugby, England. However, pressures of 1200 to 1400 lb. per sq. in. are being successfully used at Edgar Station, Boston, Massachusetts; Lakeside Station, Milwaukee, Wisconsin; and the North East Station of the Kansas City Power and Light Company. Steam pressures of 600 to 700 lb., with steam temperatures up to 750 deg. fahr., are well established for large central stations.

The steam temperature is the limiting factor [1] in modern design, due to the fact that metals now used in boilers, piping, and turbines lose decidedly in tensile and fatigue strength when the temperature exceeds 800 deg. fahr. Considerable research work is being done to determine the characteristics of metals at higher temperatures, and the temperature limit may soon be raised to 1000 deg. fahr.

Mercury may be vaporized in a special boiler at temperatures much higher than those practicable with steam. A combination mercury-steam unit of 4000-kw. capacity was installed,[2] in 1923, at the Dutch Point Station of the Hartford (Conn.) Electric Light Company. Mercury is vaporized in a special boiler at a pressure of 35-lb. gage and a temperature of 812 deg. fahr., and expanded to a 29-in. vacuum in a mercury-vapor turbine. The mercury exhaust, at a temperature of 414 deg. fahr., is condensed in a special boiler where its latent heat is used for generating steam at approximately 200-lb. gage pressure. This steam is used for operating the conventional type of steam turbine. The performance of the unit was sufficiently satisfactory to warrant the further development [3] of this type of power generation. In 1928 this company installed,[4] at the South Meadow Station, a five-stage 10,000-kw. mer-

[1] "What is the Limiting Temperature in Central Station Operation?" by V. T. Malcolm, *Mechanical Engineering*, Feb., 1928, p. 137.

[2] "The Emmet Mercury-Vapor Process," *Mechanical Engineering*, February, 1924, p. 91, and May, 1924, p. 235.

[3] "The Mercury-Vapor Power Plant at Hartford," *Power*, May 31, 1927, p. 818.

[4] *Mechanical Engineering*, January, 1928, p. 34.

cury-vapor turbine to take mercury-vapor at approximately 70-lb. pressure and 884 deg. fahr. and exhaust it at 1-lb. absolute to two mercury condensers; which are in fact steam generators producing steam at 250- to 350-lb. pressure. This steam is supplied to standard turbines.

82. Classification.—Steam boilers may be classified according to: (1) relative position of the water and hot gases, as *water-tube* and *fire-tube;* (2) arrangement of tubes, as *vertical*, *horizontal*, and *inclined;* (3) curvature of tubes, straight-tube and bent-tube; (4) location of furnace, *externally-fired* and *internally-fired;* and (5) nature of service, *stationary, locomotive,* and *marine.*

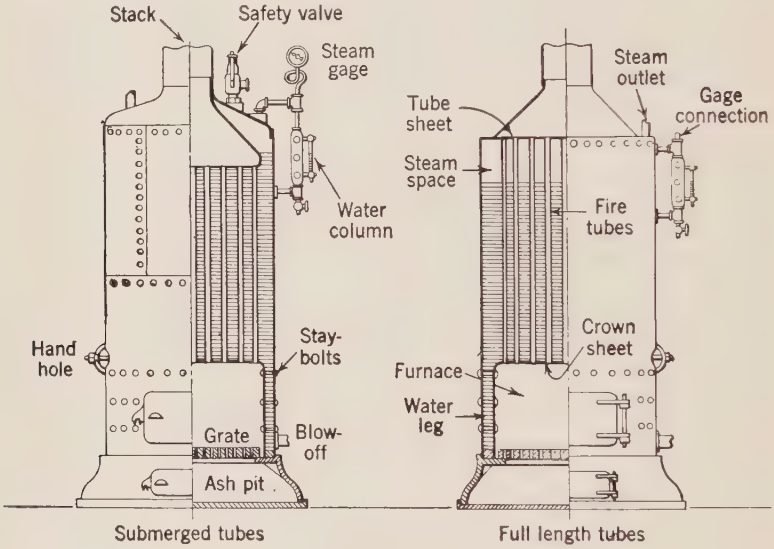


Fig. 29.—Vertical fire-tube boilers.

83. Vertical Fire-Tube Boilers.—In the fire-tube boiler the hot gases from the fire pass through the tubes. Fig. 29 shows two types of vertical fire-tube boilers, having internal furnaces and with tubes extending from the crown sheet to a tube sheet at the top of the boiler. They are built in standard sizes ranging from 20 to 48 in. in diameter and from 48 to 120 in. in length, with corresponding ratings of 3 to 50 hp.

This type of boiler is generally used where the chief requirements are: low steam pressures (less than 100 lb.), compactness, portability, low first cost, rapid steaming ability, and small quantity of steam (100 to 1500 lb. per hour). Some of the disadvantages are: small steam capacity, poor economy because of high temperature of gases entering the stack, inaccessibility for inspection and cleaning, and fluctuation of steam pressure with varying demands for steam.

Larger fire-tube boilers than those mentioned above are being built up to 500 boiler hp. Some of the disadvantages of the smaller types are minimized; so that the effects are: (*a*) improved economy because of larger grate areas and longer tubes, (*b*) suitability for steam pressures up to 200 lb. per sq. in., and (*c*) a small floor space required as compared to horizontal types of equal capacity.

84. Horizontal Return-Tubular Boilers.—In Fig. 30 is shown a longitudinal view of a return-tubular boiler and setting. The furnace consists of the combination of grates, bridge wall, fire and ash doors, the

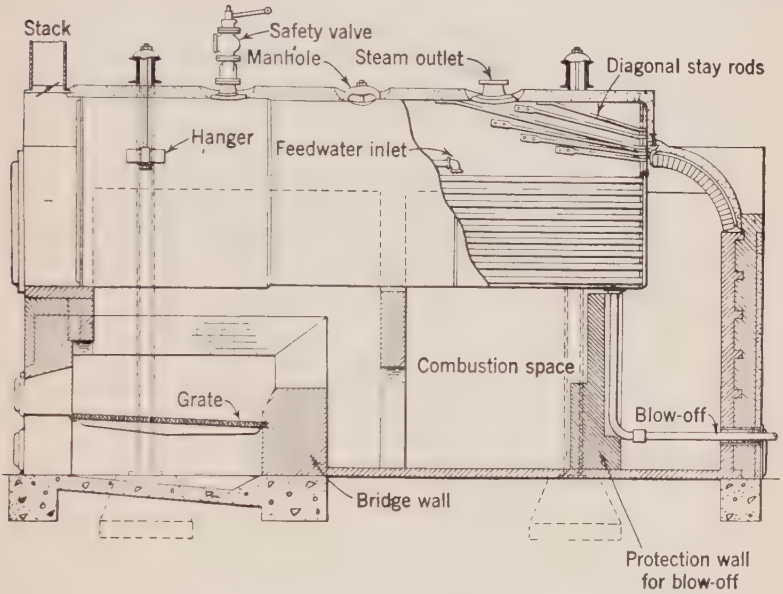


Fig. 30.—Vogt horizontal return-tubular boiler.

ashpit, and the combustion space. The front and sides of the furnace are lined with firebrick. The boiler shell is supported by steel beams and hangers; lugs resting on the brickwork may be used for small horizontal return-tubular boilers.

The hot gases pass over the bridge wall, along the bottom of the boiler shell and then through the fire tubes, toward the front of the boiler. From the front end of the tubes the products of combustion pass through the breeching into the stack.

Standard sizes of these boilers are built, in accordance with the A.S.M.E. Boiler Code, from 36 in. in diameter and 8 ft. long, to 84 in. in diameter and 20 ft. in length; ranging from 15 to 350 hp. and steam pressures up to 175 lb. They are widely used in small industrial plants

because of their compactness, low first cost, ease of cleaning, and water storage capacity. These boilers have, to some extent, the disadvantages of the vertical-tubular boilers.

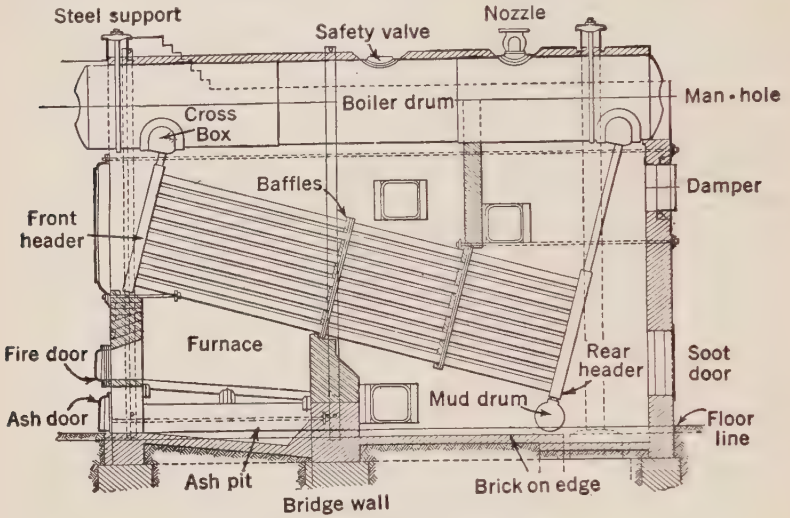


FIG. 31.—Babcock and Wilcox boiler.

85. Horizontal Water-Tube Boilers: Longitudinal Drum.—Figure 31 shows a longitudinal drum Babcock and Wilcox horizontal water-tube boiler, typical of this class. Distinguishing features of the boiler are: inclined (15 deg.) straight tubes, sectional sinuous-headers (inclined or vertical), horizontal drum or drums, and vertical (or nearly so) flue gas baffles. This boiler is built up of sections consisting of a number of tubes joined at the ends by headers; the sections are arranged side by side and connected with the steam drum by short tubes as shown. Two vertical rows of tubes are fitted to each header, the tubes being "staggered" with respect to each other, as shown in Fig. 32.

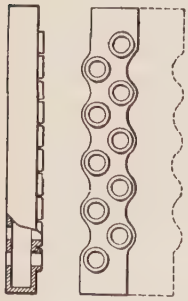


FIG. 32.—Details of sinuous header.

The boiler is suspended by stirrups from beams resting on suitable steel columns; thus the entire structure is free to expand or contract without injury to boiler or brick setting; also this makes it possible to repair the brickwork without disturbing the boiler or its piping.

The feedwater entering the front end of the drum, Fig. 33, is readily picked up by the water circulating in the boiler, and carried toward and

down the rear header; circulation is effected by the difference in density between the column of water in the rear header, and the mixed steam and water in the front headers. A deflecting steam baffle, Fig. 33, in the drum over the front headers, prevents an excess of water from being carried into the steam space. The steam passes into a perforated pipe (dry pipe) at the top of the drum and out into the steam main.

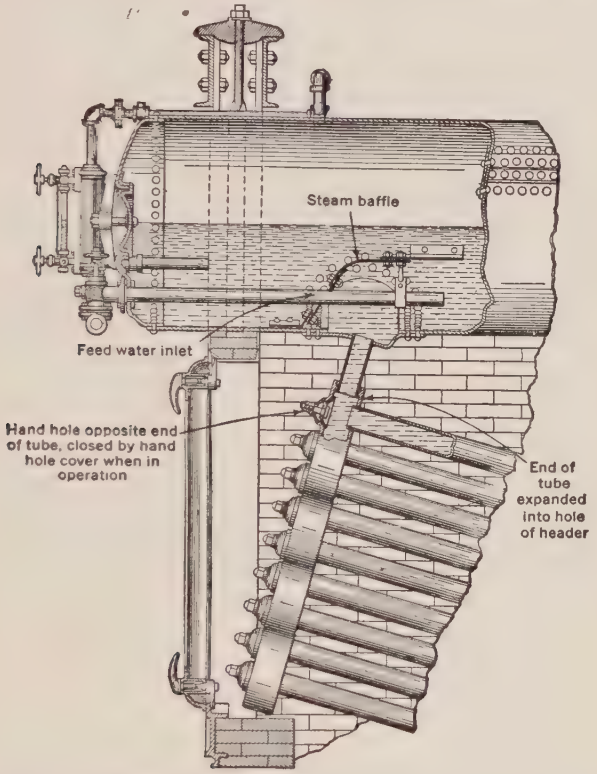

Steam baffle
Feed water inlet
Hand hole opposite end of tube, closed by hand hole cover when in operation
End of tube expanded into hole of header

Fig. 33.—Details of Babcock and Wilcox boiler construction.

Most of the impurities in the boiler water separate out and settle in the mud drum at the bottom of the rear headers, shown in Fig. 31. These impurities are removed at intervals by opening the valves in blow-off pipes leading from the mud drum. Scale, which accumulates in the tubes, is removed by mechanical tube cleaners; the latter being inserted through the hand holes in the headers opposite the ends of the tubes, Fig. 33. Scale removal from the inside of the drum is accomplished by the use of hammer and chisel, workmen entering through the manholes. Soot and ash, which accumulate on the outer surfaces of the tubes, are removed periodically by hand brush or scraper, or by a steam or air jet;

the necessary brush, scraper, or nozzle is inserted through openings in the side walls of the brickwork.

Baffle plates separate the flue gas areas around the tubes into compartments called passes. The baffles are so located that the gas passage areas are ample, and so proportioned as to give a maximum heat absorption from the gases without undue frictional resistance. In Fig. 31 the front baffle, which is made of cast iron and firebrick, is supported by the bridge wall and the tubes above the bridge wall, while the rear baffle is supported by a steel beam and the tubes. The products of combustion travel upward through the first pass across the tubes to a space beneath the drum. The hot gases are turned downward over the front baffle, across the central portion of the tubes through the second pass, and then around the lower end of the rear baffle into the third pass, and across the lower ends of the tubes to the damper and stack connection.

A section of a Heine water-tube boiler is shown in Fig. 34. It consists essentially of an inclined drum, two sheet-steel headers, inclined straight tubes connecting these headers, and horizontal baffles. A throat opening is cut into the bottom of the drum near each end, for the header (water-leg) connections.

The feedwater enters at F, at the top and near the front end of the drum, and goes into a steel mud-drum D; the latter is supported several inches above the bottom of the boiler drum. The greater part of the impurities is deposited in the mud drum and removed periodically through a blow-off pipe A. A blow-off connection B is also shown at the bottom of the rear header. The warmed water rises and flows out of the front end of the mud drum D, and there joins the circulation of the boiler water.

The products of combustion are usually directed along the tubes by the longitudinal baffles, and thence along the under side of the drum to the smoke connection. The removal of scale from the inside of the boiler tubes is accomplished in the same manner as in the Babcock and Wilcox boiler. Soot and dust are blown from the tubes by steam lance pipes inserted through the hollow staybolts S in the headers.

86. Horizontal Water-Tube Boilers: Cross Drum.—The Springfield water-tube boiler, Fig. 35, is typical of this class. This type of boiler was originally designed to meet certain conditions of headroom (as in marine service). In recent years it has become popular for central power stations and is particularly adaptable for large units (up to 2500 hp.).

The drum is located across and above the tubes, between the tops of the headers or directly above the rear header. Special features of this boiler are: sectional sinuous steel headers, vertically baffled, straight tubes, one handhole for four tubes, unrestricted water circulation, very compact, tubes staggered vertically, and no staybolts.

Fig. 67 shows a Babcock and Wilcox cross-drum boiler for high steam pressures and temperatures. It has vertical headers, and the water-heating surface is divided into two sections or decks with the superheater unit between the upper deck and lower deck of water tubes. In this boiler the drum is directly above the rear header.

87. Vertical Water-Tube Boilers: Straight-Tube Type.—The Bigelow-Hornsby boiler, Fig. 36, is made up of cylindrical units, each unit consisting of a lower and an upper drum connected by straight

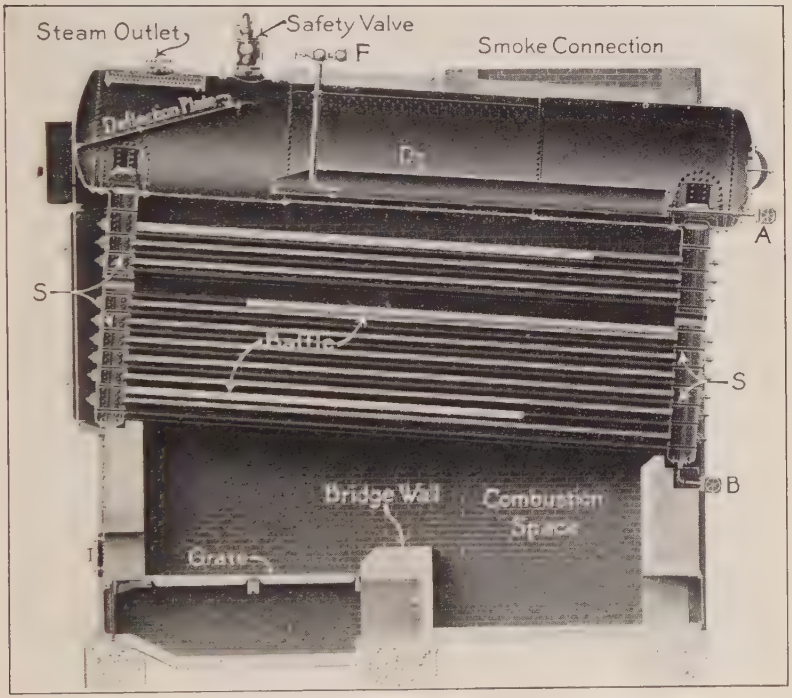


Fig. 34.—Heine H-type boiler.

tubes. Four units, as shown in Fig. 36, constitute a section, and any number of sections may be connected together. The two front units are inclined over the furnace at an angle of about 68 deg. with the horizontal; the two rear units are vertical. The upper drums are connected to a horizontal steam drum by bent tubes, as shown; the lower drums also have interconnecting tubes. Feedwater enters the top drum of the rear units, passes down the rear tubes, and then up the tubes of the front units. This boiler is built in sizes ranging from 3000 to 22,000 sq. ft. of heating

surface. The arrangement of the baffling in the Bigelow-Hornsby boiler is unusual. Fig. 37 shows a plan of the boiler units and baffles in the second, third, and fourth passes. These baffles extend along the entire length of the tubes between the units, and the gases pass over the tubes transversely. In the first pass the baffles do not extend to the top of the tubes, see Fig. 36. The baffles divide the products of combustion into thin streams passing across the tubes; thus contributing to a more

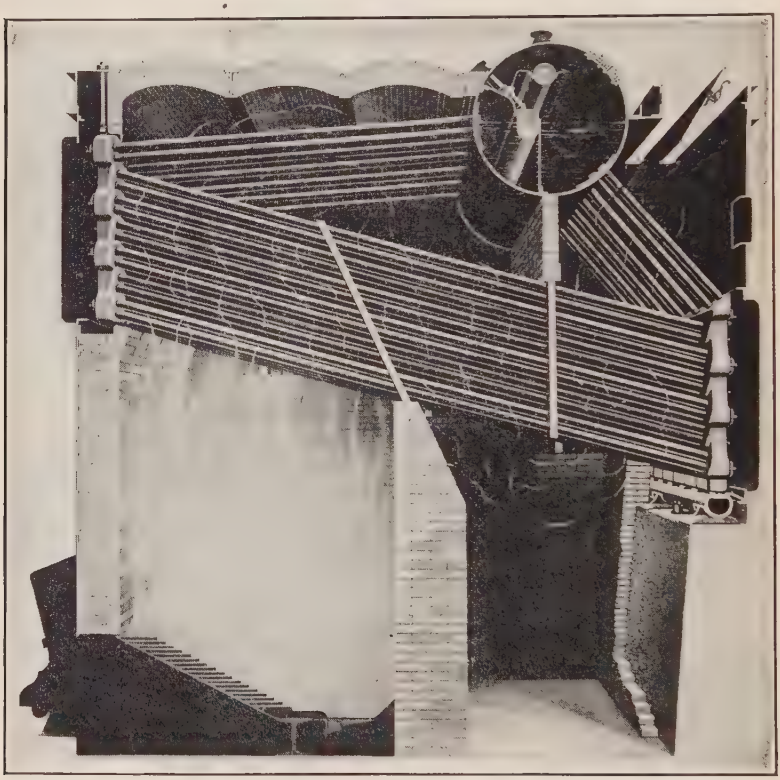


Fig. 35.—Springfield cross-drum boiler.

rapid heat transfer, than is possible when there are large unrestricted areas paralleled by the heating surface in the direction of flow of the gases.

The Edge Moor single-pass boiler, Fig. 38, consists of a number of straight-tube cylindrical units and a cross drum, connected by circulating tubes having a slight curvature near the ends. According to the manufacturer, because of the absence of baffles, the draft loss is less than half of that usually encountered for equal capacity and efficiency; exit flue gas

temperatures are low; and the boiler responds readily to high rates of evaporation without the usual loss of efficiency. Other claimed advantages are: reduced floor space, flexibility of assembly to meet unusual building conditions, and adaptability to large sizes and high steam pressures.

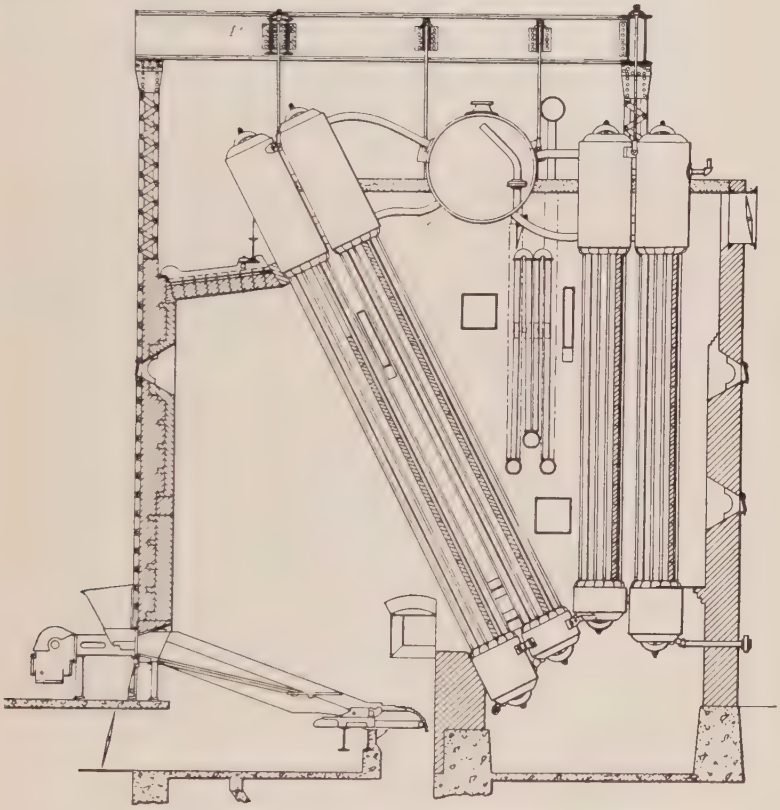


Fig. 36.—Bigelow–Hornsby boiler.

The Wickes vertical water-tube boiler, Fig. 39, consists of an upper and a lower circular drum joined by straight vertical tubes. The tubes are arranged in parallel rows, and a vertical baffle C is placed in the center extending almost to the upper drum. The furnace is located in a "Dutch oven," built out from the main boiler setting. The boiler is supported by four brackets riveted to the mud drum and resting upon plates fastened in the walls of the brickwork.

The steam drum has feed-pipe and water-column connections, also a manhole, and is normally one-third full of water; thus a large steam

space and a large disengaging surface for steam separation are provided. The hot gases pass upward over the front tubes and downward over the rear tubes to the smoke connection *DE*. The feedwater enters at the rear near the bottom of the upper drum, and flows downward in the rear

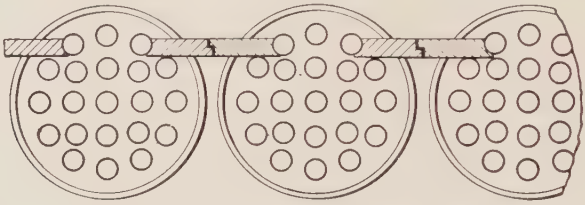


FIG. 37.—Plan of Bigelow–Hornsby boiler.

tubes and upward in the front tubes. The Wickes boiler is simple in design, requires small floor space, and is easy to inspect and clean. These boilers are built in varying sizes up to 500 hp. for steam pressures not exceeding 200 lb. per sq. in. gage.

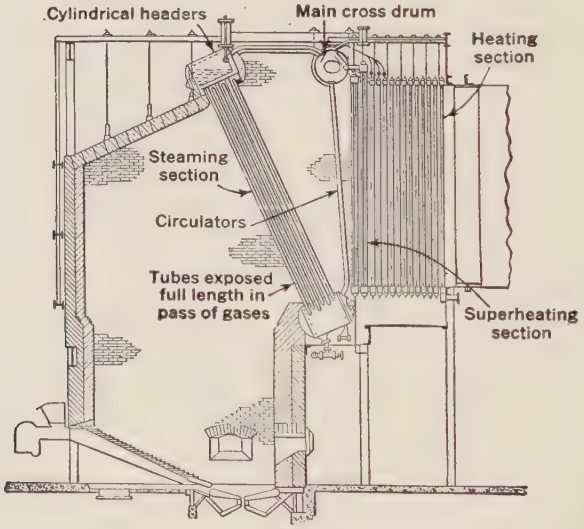


FIG. 38.—Edge Moor single-pass boiler.

88. Vertical Water-Tube Boilers: Curved-Tube Type.—Among the better-known designs of boilers, that might be named under this heading, are the Stirling, Badenhausen, Connelly, Kidwell, Ladd, and Erie City. These boilers have one or more horizontal steam drums above, connected by vertical or sloping tubes to one or more horizontal water drums

below. This type of boiler has a large number of tubes, and therefore is suitable for high capacities.

A section of the standard type of Stirling boiler is shown in Fig. 40. This boiler consists of three upper horizontal drums connected by short curved tubes; the upper drums are connected to a mud drum by three banks of long tubes so curved near the ends as to enter the drums radially.

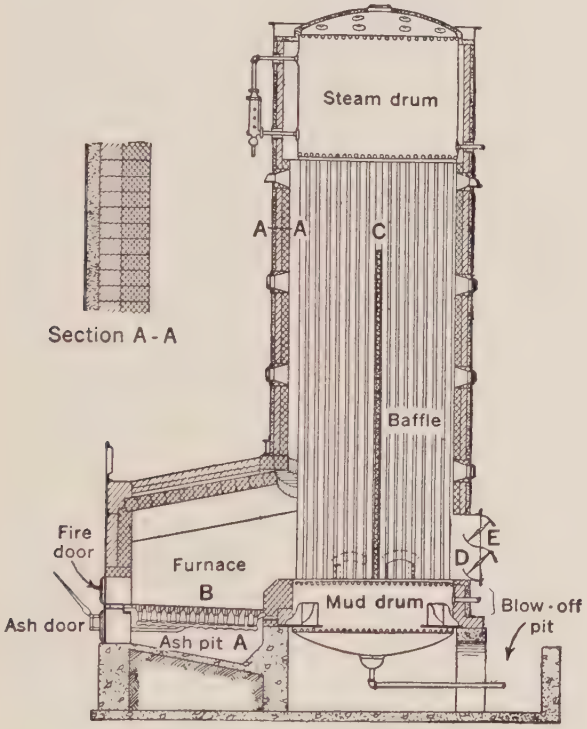


FIG. 39.—Wickes vertical water-tube boiler.

The feedwater enters the rear upper drum and is gradually heated and partly purified as it passes downward to the lower drum, where the greater part of the material precipitated by the water remains until blown off. From the lower drum the water passes up the front bank of tubes, the steam formed passes to the central drum and then to the rear drum through the upper set of tubes. The water not evaporated passes to the central drum through the lower set of tubes. This water passes from the upper central drum down the middle bank of tubes to the mud drum and returns through the front bank of tubes. Steam is taken from the rear upper drum.

The interiors of the drums are accessible, for inspection and cleaning, through manholes located in the drum ends. The tubes are cleaned by tube cleaners on the inside and by soot blowers on the outside. No handholes or staybolts are necessary.

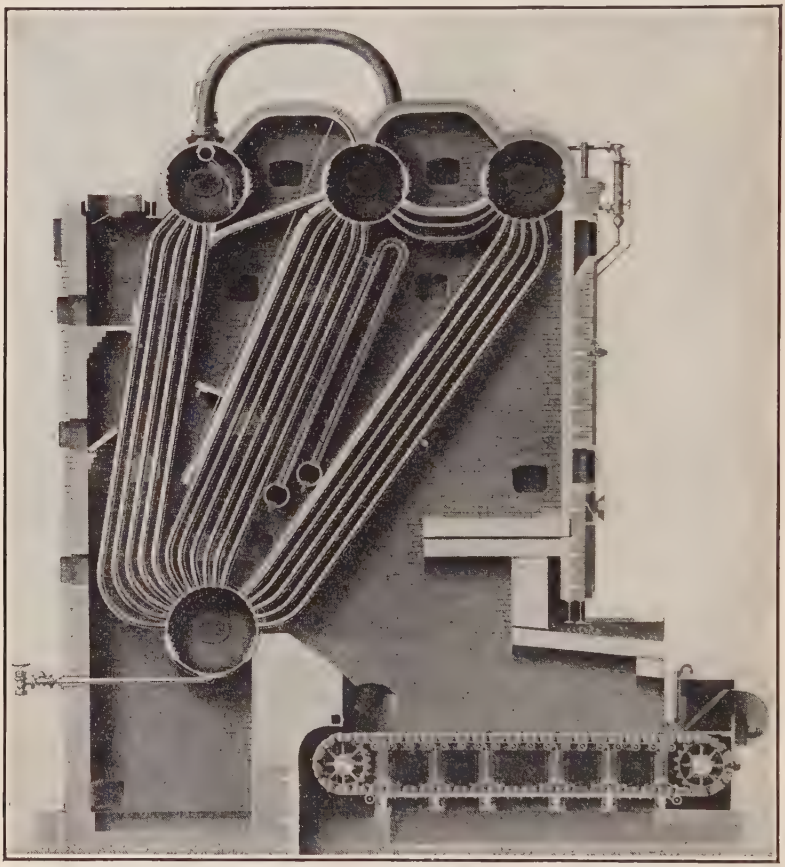


Fig. 40.—Stirling boiler, standard type.

The Badenhausen boiler, Fig. 41, the Connelly, and the Kidwell boilers are similar in design. The area of each bank of tubes is practically the same; thus excellent circulation is insured, being unrestricted and continuous at all loads. The feedwater enters the top rear drum and passes down the rear bank of tubes to the mud drum. It flows up the front bank of inclined tubes to the lower front drum, and then across the upper bank of tubes to the top rear drum. These boilers have been called " ring-flow," because of the circular path of the water as it is being

heated. The steam is taken from the upper front drum; the upper connecting tubes serve as superheating surface.

The Erie City Vertical, Fig. 42, and the small Ladd boilers in sizes up to 800 boiler hp. have two drums and three banks of tubes, as shown. The upper drum has the water level at the center line, thus affording a large steam-liberating surface. This drum is provided with a perforated feed pipe; the feedwater is passed down the rear bank of tubes at the coolest part of the boiler. A baffle plate suspended

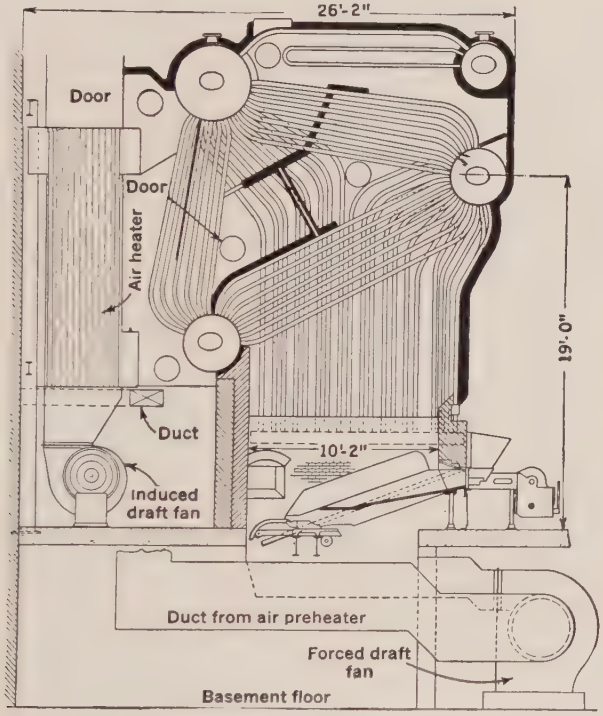


Fig. 41.—Badenhausen boiler with underfeed stoker, water-cooled walls, and air preheater.

in the drum over the front bank of tubes tends to insure an undisturbed steam space from which a perforated dry pipe collects the steam.

The lower drum is not supported from below, but by the tubes which connect it to the upper drum, the curved tubes insuring ample facility for expansion and contraction. The vertical type of boiler is particularly adaptable to waste heat installations.

In the Ladd boiler the feedwater enters a side chamber (feed box) in the lower drum and is directed upward through the rear bank of tubes, down the middle bank and then passes upward in the front bank. The

manufacturers claim that the use of this feed box in the lower drum causes the insoluble feedwater elements to be deposited near the blow-off, and prevents these particles from getting into the tubes to form scale. There is an additional blow-off connection at the bottom of the lower drum.

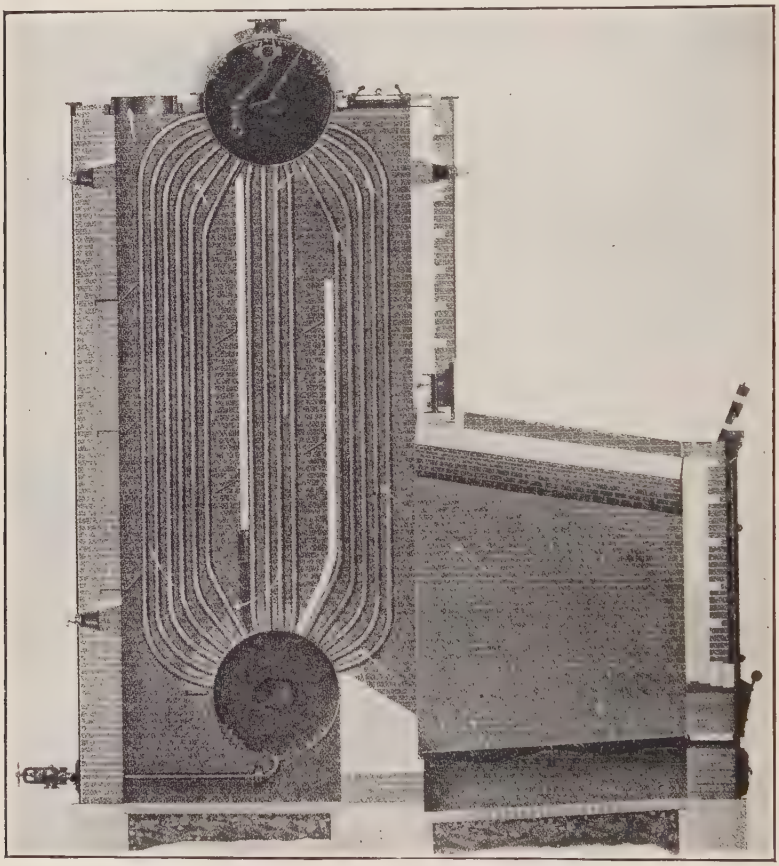


Fig. 42.—Erie City vertical boiler.

Large Ladd boilers up to 2650 hp. have been installed and engineers of the Ladd Company state that boilers of this type could be built up to 9000 hp. A number of the 2650-hp. boilers have been installed in the River Rouge plant of the Ford Motor Company, Fig. 43. They are of the double-ended type—they have four steam and water drums (two standard units) and one top steam-collecting drum S. The furnaces are designed to burn pulverized coal with burners at C, and blast furnace gas with burners at G, at the same time and in any ratio.

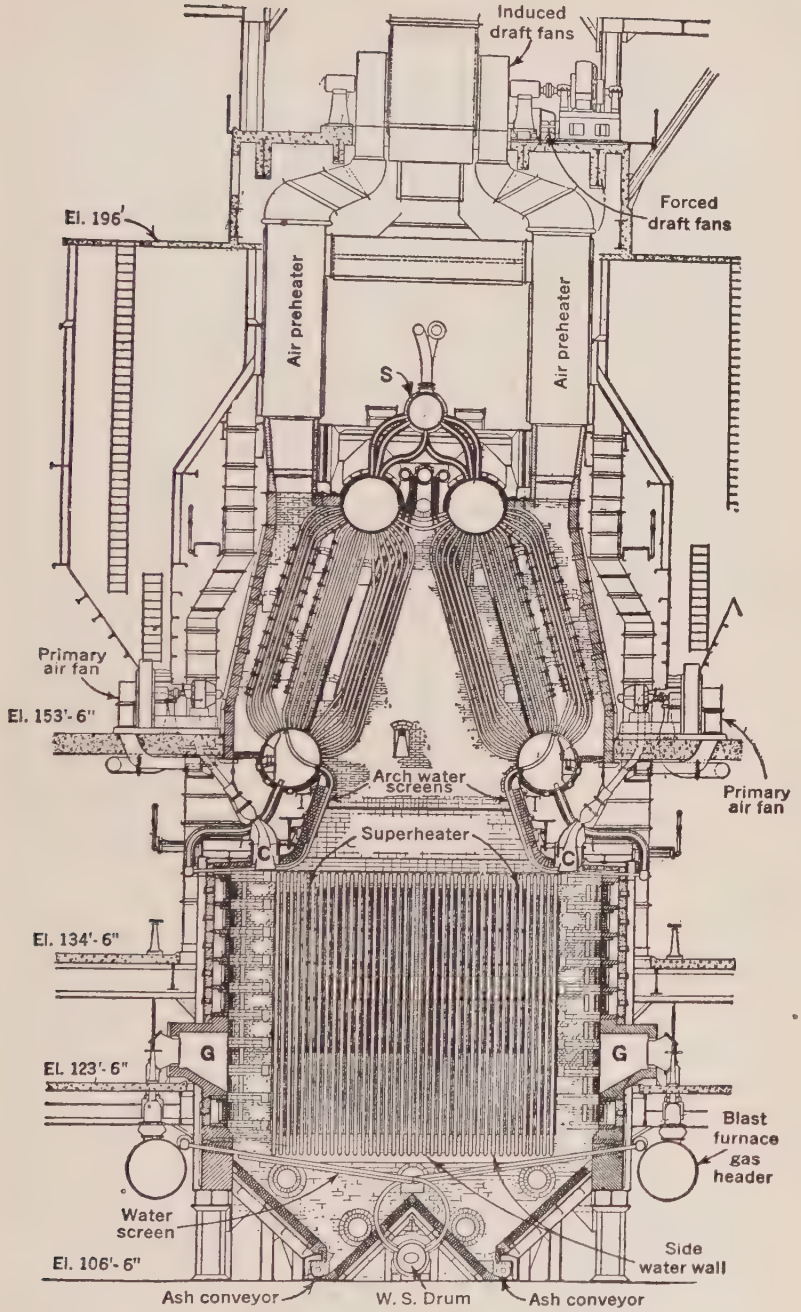


FIG. 43.—Large Ladd boiler with gas and pulverized-coal burners, water screens, radiant superheater, and air preheaters. Installed at the Fordson Plant, River Rouge, Michigan.

89. Boiler Capacity.—The American Society of Mechanical Engineers defines *boiler horsepower* as the evaporation of 34.5 lb. of water per hour from and at 212 deg. fahr. This means the conversion of 34.5 lb. of water per hour *from* water at 212 deg. fahr. into dry and saturated steam *at* 212 deg. fahr. and therefore, at atmospheric pressure (14.7 lb. abs.). Each pound of steam generated under these conditions requires the latent heat of vaporization at atmospheric pressure (14.7 lb. abs.), which is 971.7 B.t.u. The boiler hp. is, therefore, the equivalent of 33,523 B.t.u. per hour.

The weight and temperature of the water supplied to a boiler, and the pressure and quality of the steam leaving the boiler, can be obtained by test. As no power plant boilers receive feedwater at 212 deg. fahr. and convert it into steam at the same temperature, it is necessary to convert the weight of water actually evaporated per hour to the weight that would have been evaporated from and at 212 deg. fahr. and then divide by 34.5 to find the boiler hp. developed.

The *factor of evaporation* is the ratio of the heat absorbed by one pound of feedwater under actual conditions to that absorbed by one pound of water evaporated from and at 212 deg. fahr. Thus

$$F.E. = \frac{H_2 - h_1}{971.7} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

where H_2 = heat content of one pound of steam at observed pressure and quality or temperature, B.t.u. per lb. above 32 deg. fahr.

h_1 = heat content of one pound of feedwater at observed temperature, B.t.u. per lb. above 32 deg. fahr.

Equivalent evaporation per hour is the number of pounds of water which would have been evaporated per hour from and at 212 deg. fahr., if the same amount of heat had been absorbed as under given or test conditions, then

$$W_e = \frac{W(H_2 - h_1)}{971.7} = W \times (F.E.) \quad . \quad . \quad . \quad . \quad . \quad (83)$$

where W = total weight of feedwater evaporated into steam, lb. per hour at the observed pressure and quality or temperature.

The *boiler horsepower* developed may be found by using one of the following equations:

$$\text{Boiler hp.} = \frac{W(H_2 - h_1)}{971.7 \times 34.5} = \frac{W(H_2 - h_1)}{33{,}523} \quad . \quad . \quad . \quad (84)$$

$$\text{Boiler hp.} = \frac{W \times (F.E.)}{34.5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

$$\text{Boiler hp.} = \frac{W_e}{34.5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (86)$$

The *rated horsepower* of a boiler is dependent upon the water-heating surface. The heating surface is the total surface of the boiler shell and tubes exposed to hot gases, having water on the opposite side of shell and tubes. Recent practice rates all stationary boilers on a basis of 10 sq. ft. of heating surface per boiler hp. By this method, boilers of equal heating surface are given the same rating, but one boiler may have its heating surface arranged more advantageously than another and thus have a greater capacity for like operating conditions. Most power plant boilers can develop from 300 to 500 per cent of their normal rating and many are operating for long periods at 250 per cent rating. The efficiency of the boiler may be decreased when it develops much more than its rated capacity. This decrease in efficiency is generally not appreciable until the boiler is from 100 to 200 per cent or more overloaded, the decrease being due to the inability of the boiler to absorb the extra heat of the fire.

90. Boiler Performance.—The performance of a boiler may be expressed in terms of: efficiency, rate of combustion, evaporation per pound of fuel, number of B.t.u. absorbed per sq. ft. of heating surface per hour, boiler hp. and percentage of rating developed.

The overall *boiler efficiency* is the ratio of the heat absorbed to that supplied, i.e., output divided by input, or

$$e_b = \frac{W(H_2 - h_1)}{W_f \times F} \times 100 \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

in which e_b = efficiency of boiler, furnace and grate in per cent.
W_f = total weight of fuel fired per hour, lb.
F = higher heating value, B.t.u. per pound of fuel, as fired.

Example.—A boiler evaporates 40,000 lb. of water per hour from a feed-water temperature of 218 deg. fahr. to steam at 180 lb. per sq. in. (absolute pressure) and a quality of 97 per cent. Weight of coal fired per hour, 4500 lb., higher heating value of the coal as fired 11,800 B.t.u. per lb. Required the factor of evaporation, equivalent evaporation per hour, boiler hp. developed and the efficiency of boiler, furnace and grate.

Solution.—Substituting in (82)

$$F.E. = \frac{345.2 + (0.97 \times 852.0) - (218 - 32)}{971.7}$$

$$F.E. = \frac{985.64}{971.7} = 1.014$$

Substituting in (83)

$$W_e = \frac{40{,}000\,[345.2 + (0.97 \times 852.0) - (218 - 32)]}{971.7}$$

$$W_e = 40{,}000 \times 1.014 = 40{,}560 \text{ lb. per hr.}$$

Substituting in (86)

$$\text{Boiler hp.} = \frac{40{,}560}{34.5} = 1176.$$

Substituting in (87)

$$e_b = \frac{40{,}000\,[345.2 + (0.97 \times 852.0) - (218 - 32)]}{4500 \times 11{,}800} \times 100$$

$$e_b = \frac{3943}{5310} \times 100 = 74.3 \text{ per cent.}$$

The rate of combustion is expressed in pounds of fuel burned (*a*) per sq. ft. of projected grate surface per hour, or (*b*) per cu. ft. of furnace volume per hour. The boiler hp. that can be developed is dependent on the rate of combustion, i.e., upon the kind of fuel, size and kind of grate and quantity of air supplied. Rates of combustion as high as 200 lb. of coal per sq. ft. of grate surface per hour have been obtained in locomotive boilers with a resultant high capacity, but a lowered efficiency. In modern steam power plants, the rate of combustion for a given boiler setting for maximum efficiency is determined experimentally. The efficiency increases with the capacity (boiler hp.) and rate of combustion until a maximum is reached; beyond this point the efficiency decreases with increase of capacity, see Figs. 44 and 45. In Fig. 44 the the efficiency peak occurs between 125 and 225 per cent rating, beyond which the efficiency decreases because of the heat lost in increased flue gas temperature and an increase in the combustible (unburned carbon) in the refuse. A boiler having a pulverized coal furnace and an economizer showed an efficiency peak, Fig. 45, between 300 and 400 per cent of boiler rating. Comparison of the flue gas temperature curves shows how the use of an economizer decreases the heat carried away by the flue gases.

Small hand-fired boiler equipment may operate with overall efficiencies varying from 40 to 65 per cent. Stoker-fired boilers may show from 60 to 80 per cent efficiency, depending upon the size of the plant, and the amount of heat-saving equipment in operation. Large pulverized coal installations have shown boiler efficiencies of 80 to 85 per cent.

If air preheaters or economizers are in use in such plants, overall efficiencies of 82 to 88 per cent may be obtained at times. Large oil-burning boiler plants show efficiencies between those of stoker-fired and pulverized coal equipment. It should be noted that the average all-year boiler-operating efficiencies are well below the high efficiencies obtained for short periods of time during test conditions. In steam

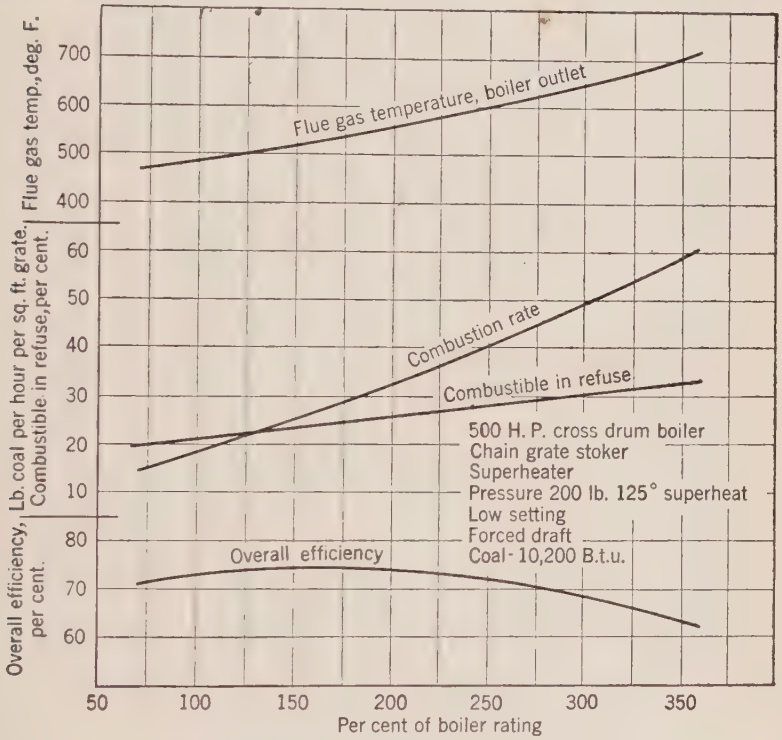


FIG. 44.—Typical boiler performance curves.

boiler practice the heat not utilized per pound of fuel varies from 12 to 60 per cent of the calorific value of the fuel.

91. Boiler Heat Losses.—The heat losses usually considered and most readily calculated incident to the combustion of fuels in a boiler furnace are:

1. Heat carried away in the dry flue gases.
2. Evaporation and superheating of moisture in the fuel.
3. Evaporation and superheating of moisture formed by burning the hydrogen of the fuel.
4. Superheating of water vapor in the air supplied.

5. Formation of carbon monoxide.
6. Unconsumed hydrogen-carbon compounds in the dry flue gases.
7. Unconsumed carbon in the ash and refuse.
8. Radiation and unaccounted-for losses.

A tabulation of the various heat losses, including the heat absorbed by the boiler, constitutes the boiler and furnace *heat balance*. In this

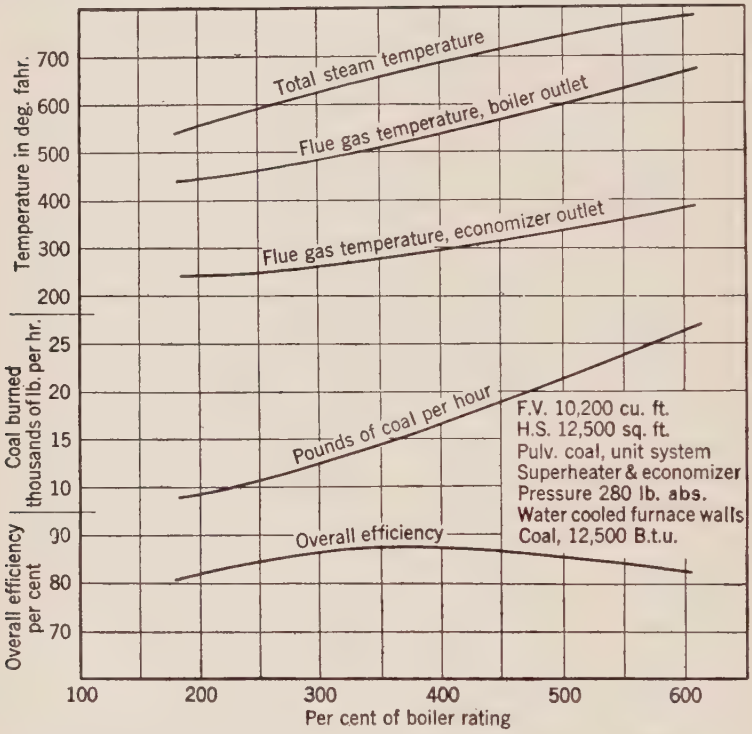


Fig. 45.—Performance of boiler No. 15; C. R. Huntley Station, Buffalo, N. Y.

way the distribution of heat per pound of fuel can be obtained and the relative proportions of the various losses studied.

1. *Heat carried away in the dry flue gases.*—The heat lost in the dry flue gases is usually the largest loss and may be calculated by the following equation,

$$h_1 = W_{dg} c_p (t_g - t_r) \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

in which h_1 = B.t.u. lost per pound of fuel.
W_{dg} = weight of dry flue gas per pound of fuel.
c_p = mean specific heat of dry gas (about 0.24).

t_g = temperature of escaping gases, deg. fahr.
t_r = temperature of air entering furnace, deg. fahr.

2. *Evaporation and superheating of moisture in the fuel.*—The moisture in the fuel is evaporated and leaves the boiler setting in the form of superheated steam.

$$h_2 = W_m[212 - t_r + 971.7 + 0.47(t_g - 212)]$$

or

$$h_2 = W_m(1084.1 - t_r + 0.47t_g) \qquad (89)$$

in which h_2 = B.t.u. lost per pound of fuel.
W_m = weight of free moisture per pound of fuel.

3. *Evaporation and superheating of moisture formed by burning hydrogen of the fuel.*—The hydrogen in the fuel burns to water and the latter leaves the boiler setting in the form of superheated steam.

$$h_3 = 9H(1084.1 - t_r + 0.47t_g) \qquad (90)$$

in which h_3 = B.t.u. lost per pound of fuel.
H = weight of hydrogen per pound of fuel.

4. *Superheating of water vapor in the air supplied.*—This heat loss is small and may be computed thus:

$$h_4 = W_v(t_g - t_r)0.47 \qquad (91)$$

in which h_4 = B.t.u. lost per pound of fuel.
W_v = weight of water vapor supplied with air per pound of fuel.
W_v = relative humidity × weight per cubic foot of water vapor (from Table XVII) at t_r × volume per pound of dry air at t_r × weight of dry air per pound of fuel.

5. *Formation of carbon monoxide.*—This heat loss is usually small and is caused by an insufficient air supply, some of the carbon in the fuel being burned to carbon monoxide.

$$h_5 = \frac{CO}{CO_2 + CO}(14600 - 4440)C_1$$

or

$$h_5 = \frac{10160\,CO}{CO_2 + CO} \times C_1 \qquad (92)$$

in which h_5 = B.t.u. lost per pound of fuel.
CO_2 and CO = percentages by volume as determined by the flue gas analysis.
C_1 = weight of carbon actually burned per pound of fuel.

6. *Unconsumed hydrogen-carbon compounds in the dry flue gases.*—Hydrogen-carbon compounds are distilled from the fuel at furnace temperatures and unless sufficient air is supplied, these hydrocarbons may escape unburned or be decomposed forming soot, H_2 and CO. The unconsumed hydrocarbons in the flue gases may be determined by a complete gas analysis, and the heat loss h_6 calculated by Dulong's formula, equation 67.

7. *Unconsumed carbon in the refuse.*—Some of the carbon in the fuel, either unburned or partially burned, may fall into the ashpit. This loss depends upon the type of grate, rate of combustion and size and kind of coal.

$$h_7 = \frac{14{,}600\, W_r C_r}{W_f} \quad . \quad . \quad . \quad (93)$$

in which h_7 = B.t.u. lost per pound of fuel.
W_r = weight of refuse, lb. per hour.
C_r = weight of carbon per pound of refuse.
W_f = weight of fuel as fired, lb. per hour.

8. *Radiation and unaccounted-for heat losses.*—These losses are determined by subtracting the heat absorbed by the boiler and the heat losses 1 to 7 inclusive from the heat value per pound of fuel. Unaccounted-for heat losses are those which cannot be measured.

h_8 = B.t.u. per pound of fuel − B.t.u. absorbed by boiler per pound of fuel − $(h_1 + h_2 + h_3 + h_4 + h_5 + h_6 + h_7)$. . . (94)

Example.—Tabulate the various heat losses and show method of calculation from the following data:

Heat absorbed by boiler, 74.8 per cent of the heating value of the fuel.

Analysis of coal as fired, per cent by weight:

Carbon	62.0	Nitrogen	1.0
Hydrogen	4.0	Free moisture	8.0
Oxygen	8.0	Ash	15.0
Sulphur	2.0		
		Total	100.0

Heating value of coal as fired, 11,800 B.t.u. per lb.

Flue gas analysis, per cent by volume:

CO_2	13.0	O_2	6.0
CO	1.0	N_2 (by difference)	80.0

Temperature of air entering furnace, 74 deg. fahr.; temperature of flue gases, 452 deg. fahr.; temperature of steam in boiler, 360 deg. fahr.; relative

humidity of air entering furnace, 70 per cent; coal used per hour, 2000 lb.; refuse per hour, 420 lb.; and unburned carbon in refuse 18 per cent.

Solution.—The actual weight of carbon appearing in the flue gas from equation 76 is

$$C_1 = \frac{(2000 \times 62) - (420 \times 18)}{2000 \times 100} = 0.5822 \text{ lb. per lb. of coal.}$$

Weight of dry air supplied, equation 77,

$$W_{aa} = \left[\frac{28 \times 80}{12(13 + 1)0.769}\right] \times 0.5822 = 10.10 \text{ lb. per lb. of coal.}$$

Weight of dry chimney gas, equation 80,

$$W_{dg} = 0.5822\left[\frac{(4 \times 13) + 6 + 700}{3(13 + 1)}\right] = 10.51 \text{ lb. per lb. of coal.}$$

Heat Losses per Pound of Coal as Fired

Name of Heat Loss	Method of Calculation	B.t.u.	Per Cent
Heat absorbed by boiler	0.748 × 11,800	8,826	74.80
Dry flue gases	10.51 × 0.24(452–74)	954	8.08
Moisture in fuel	0.08[(1084.1–74 + (0.47 × 452)]	98	0.83
Hydrogen in fuel	9 × 0.04[(1084.1–74 + (0.47 × 452)]	440	3.73
Water vapor in air	0.70 × 0.00131 × 13.45 × 10.10(452–74)0.47	22	0.18
Incomplete combustion	[(10,160 × 0.01) ÷ (0.13 + 0.01)]0.5822	422	3.58
Carbon in refuse	(14,600 × 420 × 0.18) ÷ 2000	552	4.68
Radiation, etc	By difference	486	4.12
	Total	11,800	100.00

92. Boiler Accessories.—All devices which are directly connected with a boiler or which facilitate its operation and maintenance are called *boiler accessories*. The accessories usually provided are: steam pressure gage, water column, feedwater regulator, safety valve, fusible plug, blow-offs, mechanical tube cleaner and soot blower.

The steam pressure in a boiler is usually indicated by a pressure gage. Pressure gages in general use are of two classes, known as the Bourdon and diaphragm gages. In the Bourdon gage, see Fig. 46, the pressure is exerted on the interior of a tube, oval in cross-section, bent to fit into a circular case. The application of pressure tends to make the cross-section round and thus to unroll or straighten the tube; this motion, communicated by means of a sector, levers and gears, rotates a hand above a calibrated dial and the pressure, in lb. per sq. in. above that of the

atmosphere, may be read. In the diaphragm gage the pressure is resisted by a corrugated disk or diaphragm. The motion is transmitted to the hand in the same manner as in the Bourdon gage. These gages (Bourdon and diaphragm) are affected by heat and hence should have a bent tube or equivalent containing water, between the gage and the steam space. These protecting devices are called siphons.

The water column and attached gage glass, Fig. 47, are connected to the front of the boiler drum, so that the water level may be visible from the boiler-room floor. The top of the water column is connected with the steam space and the bottom of the water column to the water space in the boiler as shown. Mud and sediment are removed by blow-off cocks attached to the lower end of the water column and gage glass. Gage cocks (not shown in Fig. 47) are used as a check on the indicated water level. The water column shown is provided with high and low water safety alarms. Float L holds the valve lever up and the alarm valve closed as long as the water level is above the lower gage cock. When the level falls, L opens the valve and the released steam blows the whistle or operates an electric alarm. The high water alarm has a valve to the signal which is kept closed by the weight of float H as long as the water is below the upper gage cock. The water level in some horizontal water-tube boilers is maintained at a point where the drum is about half-full of water; the water column is usually connected so that the gage glass is half filled at the desired water level.

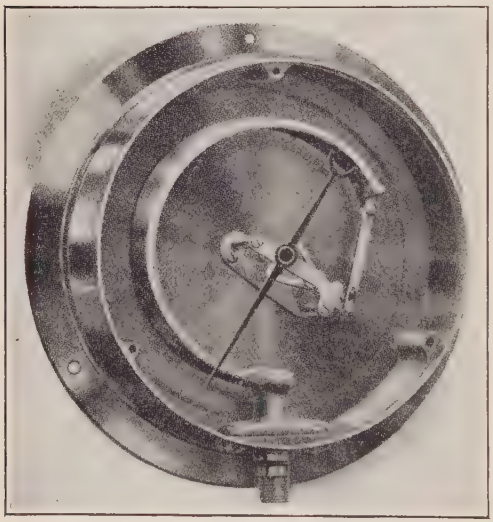


Fig. 46.—Ashcroft Bourdon pressure gage.

The purpose of a feedwater regulator is automatically to supply a boiler with the same amount of water that is being converted into steam, thereby maintaining a constant water level.

The following types of feedwater regulators are in common use: float-lever, thermo-expansion, and thermo-pressure. They are generally used in connection with suitable pump governors. In the float-lever type the actuating member is a float which rises and falls with the water

level. When the water level falls, Fig. 48, the movement of the float *B* opens the feedwater control valve *F* through levers *EE* and more feedwater enters the boiler. *A* is connected to the boiler steam space and *D* to the water space. The float *B* is non-collapsible, open at the top and kept filled with water by condensate which is continually forming in the

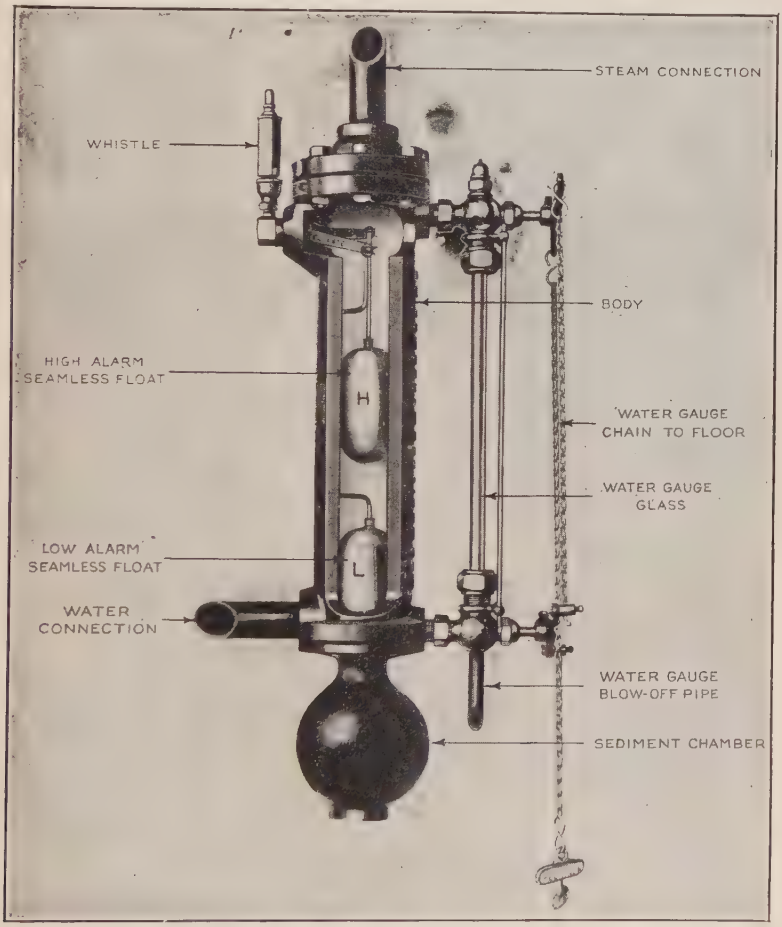


Fig. 47.—Reliance water column and safety alarms.

steam space above and falling into the small pan. *H* is a counterweight which gives the water-filled float its buoyancy and keeps the center line of the float coincident with the water level. Water gage connections may be made at *C*. *G* is an alloy metal guide with enough clearance to admit a small amount of cooling feedwater, thus the water in the float chamber is kept below the temperature of the steam and evaporation

is prevented. The removal of cover plate *I* gives access to working parts of the regulator.

In the thermo-expansion type of feedwater regulator, the expansions and contractions of an inclined metal tube are produced by the fall or rise in the water level. As Fig. 49 shows, the water level in the inclined thermostat tube *T* is the same as the water level in the boiler *B*. With the boiler water at normal level 2, half of the expansion tube is filled with steam and half with water. When the steam output of the boiler increases, the water in the boiler *B* and tube *T* begins to fall to position 1, more steam is now contained in the tube; *T* then expands and opens the control valve *V* through lever *L*, thus increasing the rate of feedwater flow. At position 3 the water level is high, tube *T* contracts, thus closing valve *V*.

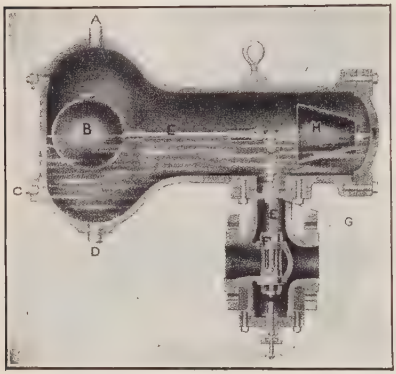


FIG. 48.—Stets float type feedwater regulator.

Safety valves are used to protect boilers against excessive steam pressures, by automatically opening at a desired pressure and allowing the

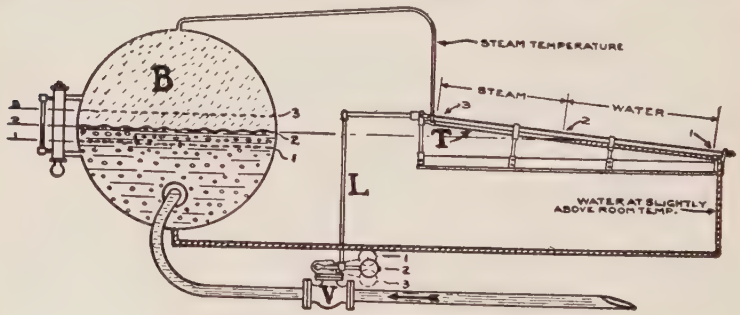


FIG. 49.—Copes thermostatic feedwater regulator.

steam to escape. Fig. 50 shows a section through a Pop Safety Valve (the only type sanctioned by the A.S.M.E. Boiler Code) in which the boiler pressure is resisted by a spring. The spring tension may be adjusted at the top and a lever is provided for manually testing the valve daily. The number and size of safety valves for a given boiler are usually specified by city or state legislation or by insurance companies.

The fusible plugs, Fig. 51, are used to protect the boiler. They are

inserted at the lowest permissible water level. They consist of steel or bronze bushings, filled with a tin alloy, screwed into the boiler (location as listed by the A.S.M.E. Boiler Code). The melting point of the alloy is above the temperature of the steam and below the temperature of the hot gases. When the water level is low enough to uncover the plug, the alloy melts and steam escapes. In this way probable excessive boiler pressure is prevented and the escaping steam attracts the attention of the attendant, so that precautions may be taken to prevent overheating of the boiler metal.

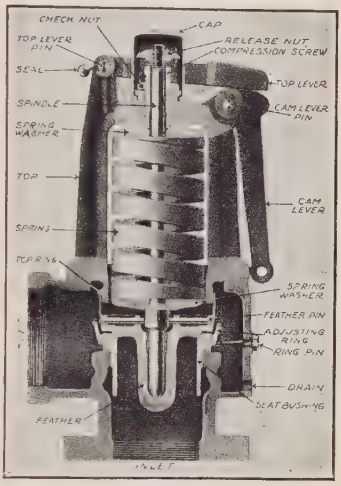


Fig. 50.—Consolidated pop safety valve.

The blow-off connections, as shown in the boiler illustrations of Figs. 29 to 42, made at the lowest part of the boiler and sometimes also at the water level (called surface blow-off) are used for the removal of water to carry off mud, sediment, scale, scum, etc. Impurities from the large quantities of water evaporated are constantly being precipitated due to the high temperatures in boilers. A continuous (bottom) blow-off using a small pipe is sometimes used to remove the impurities as they are precipitated; however, when this is done the larger blow-off valves must be opened occasionally for a more thorough removal of mud, etc.

Feedwater impurities combine mechanically and chemically during boiler operation, to form a layer of scale on the heating surfaces. The methods used to decrease the formation of scale by purification of the feedwater are discussed in Chapter IX. The heat loss due to boiler scale varies with the temperature and rate of gas and water flow. Experiments[1] have shown a heat conductivity loss of about 10 per cent for $\frac{1}{16}$-in. thickness of scale, for boiler tubes (lower rows) near the furnace. Scale may be the cause of localized overheating, resulting in serious burning, blistering and damaging of tubes and plates. All boilers require the use of a mechanical tube cleaner at definite intervals in order to remove

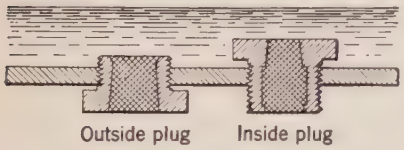


Fig. 51.—Typical fusible plugs.

[1] "Heat Transmission Through Boiler Tubes," by H. O. Croft, University of Illinois Engineering Experiment Station Bulletin No. 168, 1927.

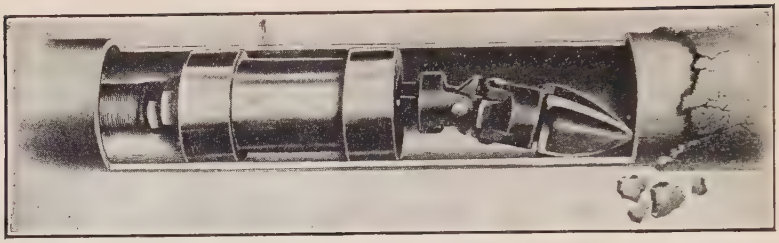


FIG. 52.—Liberty hammer-type tube cleaner.

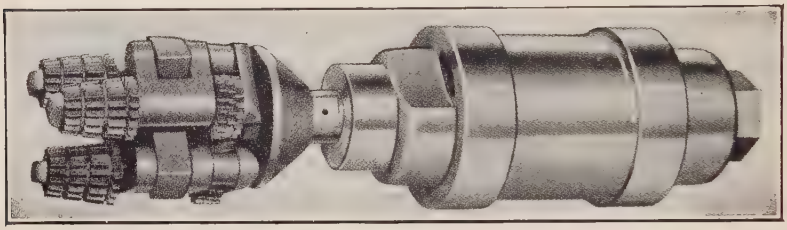


FIG. 53.—Lagonda revolving-cutter type tube cleaner.

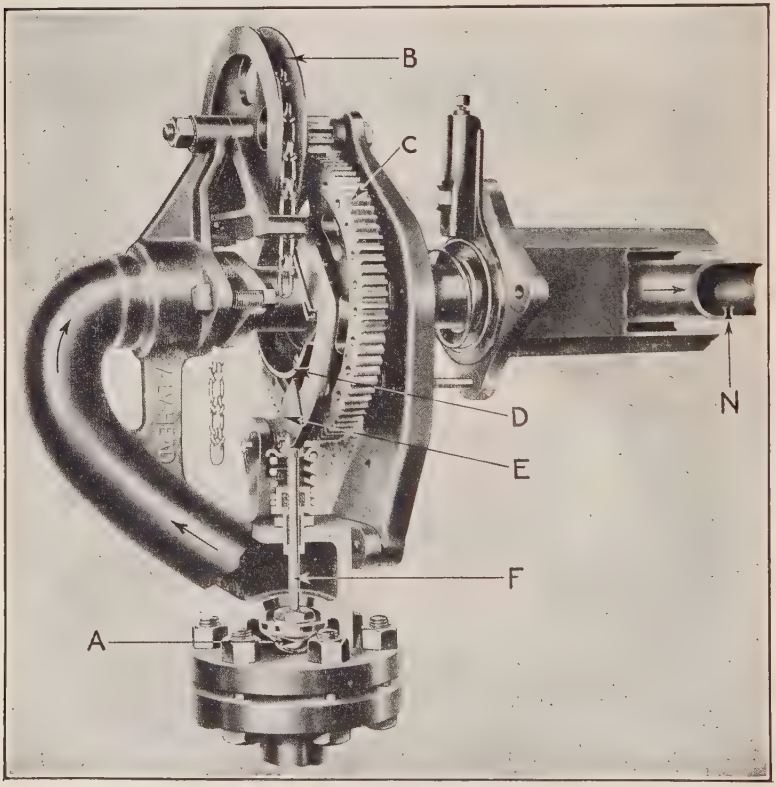


FIG. 54.—Diamond soot blower.

effectively the scale formed on the tube surfaces. Mechanical tube cleaners may be divided into two classes: (1) The *hammer* type, Fig. 52, which loosens the scale by rapid blows, is applicable to water and fire-tube boilers. (2) The *revolving cutter* type, Fig. 53, removes the scale by rotating cutting tools and is used in water tubes only. The most popular type of tube cleaner to-day is the air-driven cleaner (60 to 90 lb. pressure), because it is possible to obtain greater power and speed than when steam or water under pressure is used.

Soot is an excellent heat-insulating material and hence its removal from boiler heating surfaces is desirable. Soot is broadly understood to include all the solid matter leaving the fuel bed with the gaseous products of combustion. Boiler surfaces exposed to the flue gases are usually freed from ash and soot by hand-operated steam lances, mechanical soot blowers, brushes or similar devices. Mechanical soot blowers, Fig. 54, are permanently installed in boilers, as shown in Fig. 55, and so located that all parts of the heating surface subjected to soot deposit may be cleaned with jets of steam, air, or a mixture of air and steam. With the soot blower shown in Fig. 54, pulling the operating chain automatically opens valve A and revolves the blower element N. Sheave B rotates gear C, and valve A is opened through cam D and trigger E; steam then flows through the blower head to the nozzles at N, as indicated by the arrows. The soot blower is designed to rotate in one direction, sweeping over the blowing arc once every revolution, and when the blowing arc is completed, the steam flow is shut off by cam D. The operator may revolve the nozzle through any number of blowing arcs required to effect the cleaning.

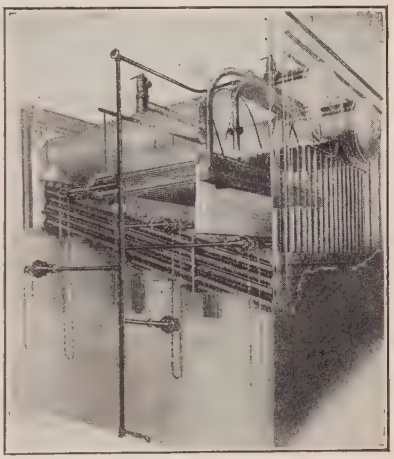


FIG. 55.—Typical installation of mechanical soot blowers.

PROBLEMS

1. A boiler generates dry steam at 150 lb. per sq. in. gage. Barometer 28.60 in., feedwater 180 deg. fahr. and water evaporated per hour is 40,000 lb. Required the factor of evaporation, equivalent evaporation per hour and the boiler hp. developed.

2. A water-tube boiler delivers 25,000 lb. of superheated steam per hour at 300 lb. per sq. in. absolute pressure and a temperature of 700 deg. fahr. (*a*) Find

the degrees of superheat. (*b*) If the steam leaves the boiler containing 2 per cent moisture, how many boiler hp. are developed in the superheater?

3. A boiler test showed an actual evaporation of 8 lb. of water per lb. of coal as fired. The feedwater temperature was 168 deg. fahr., steam pressure 200 lb. per sq. in. absolute and superheat 100 deg. fahr. Find the overall boiler efficiency, if the calorific value of the coal as fired was 12,000 B.t.u. per pound.

4. The following data were obtained from a boiler test: Water fed to boiler 24,700 lb. per hour; feedwater temperature 62 deg. fahr.; steam pressure 180 lb. gage; quality of steam 97 per cent and barometer 29.21 in. (*a*) Find the boiler hp. developed. (*b*) Determine the number of square feet of heating surface if the boiler was developing 150 per cent of its rating during this test.

5. A boiler has an overall efficiency of 70 per cent. Coal is burned at the rate of 500 lb. per hour. Moisture in coal is 6 per cent and the heating value per pound of dry coal is 12,000 B.t.u. Boiler pressure is 150 lb. per sq. in. absolute and steam discharged is at a temperature of 400 deg. fahr., feedwater temperature 202 deg. fahr. (*a*) How many pounds of steam are discharged per hour? (*b*) Find the boiler hp. developed. (*c*) Determine the equivalent evaporation, pounds per pound of dry coal. (*d*) If coal is purchased at $4.00 per ton, find the coal cost per thousand pounds of equivalent evaporation.

6. A boiler discharges 8000 lb. of steam per hour into the steam main at 99 per cent quality. Feedwater temperature 185 deg. fahr. Boiler pressure 165.5 lb. per sq. in. gage, barometer 29.58 in. (*a*) What is the boiler hp. developed? (*b*) The calorific value of the coal as fired is 10,500 B.t.u. per pound. Efficiency of boiler, furnace and grate is 60 per cent. How many pounds of coal are required per hour?

7. The following data were obtained during a test on a water-tube boiler: Heating surface 5080 sq. ft.; water fed to boiler 29,200 lb. per hr.; coal used 4300 lb. per hr.; dry refuse from ashpit 840 lb. per hr.; and combustible in dry refuse 22 per cent. Feedwater temperature 144 deg. fahr.; boiler room temperature 62 deg. fahr.; relative humidity of air 55 per cent; flue gas temperature 510 deg. fahr. Steam pressure 120 lb. gage; dry steam and barometer 29.39 in. Coal analysis, per cent by weight, as fired: carbon 65; hydrogen 3, nitrogen 1, oxygen 7; moisture 13; ash 11; and B.t.u. per lb. 10,900. Flue gas analysis, per cent by volume: CO_2 14; CO 0.8; O_2 5.5; and N_2 79.7 (by difference). Tabulate the various heat losses per pound of coal as fired and show method of calculation for each item.

CHAPTER VIII

BOILER AUXILIARIES

93. Boiler Settings and Furnaces.—The enclosing and supporting structure for a boiler is called the *boiler setting*. Boilers above 100-hp. rating are usually supported by steelwork independent of the brickwork, Figs. 30 and 31. That part of the boiler setting where the fuel is burned is called the *furnace*. The generation of steam may be divided into two distinct operations, (1) the burning of the fuel, and (2) the heat absorption by the water within the boiler. Efficient utilization of the fuel is dependent to a large extent upon the correct design and construction of the furnace and boiler setting, as has been explained in Article 64, Chapter V.

The outer walls of a brick setting are usually built of hard red brick while the inner surface is faced with high-grade firebrick capable of withstanding high temperatures. Extremely high furnace temperatures are the rule to-day in the operation of large boilers and, consequently, firebrick must be of a quality that will stand up under severe conditions. However, to prolong the usefulness of the furnace lining, perforated brick blocks for air admission, radiant types of superheaters (*R*, Fig. 72), or water-circulating pipes (Fig. 67) have been installed in furnace walls.

The boilers shown in Fig. 29 are said to be *internally-fired* because the furnace is enclosed in the lower part of the boiler shell, and surrounded by an annular water-leg. In an *externally-fired* boiler the furnace is separate from the boiler, i.e., the fire is outside the boiler shell, as in Fig. 30. Boiler settings also include, the foundation, ashpit, inspection doors, metal boiler fronts (opposite the headers), baffles, furnace arches and heat-insulating covering for the tops.

The Dutch-oven furnace, Figs. 39 and 42, is located in an extension of the boiler setting. Due to hot gases having a greater distance to travel before coming in contact with the cooler boiler surfaces, more time is allowed for complete combustion. The roof of the furnace absorbs heat from the burning coal, becomes white hot, and radiates heat back upon the coal and in this way aids combustion.

94. Grates.—A *grate* is a sectional metallic structure designed to support the fuel in a furnace, and is so made that air for combustion can

pass through the openings to the fuel. Most grates are made up of sections of cast iron; cast iron is cheap and is less liable to be destroyed at high temperatures. The openings or air spaces depend upon the kind of fuel used, and the combined area of the openings varies from 20 to 40 per cent of the total grate area. The total grate area depends upon the heating value of the coal, and the rate of combustion. Manufacturers usually provide about 1 sq. ft. of grate surface for every 40 to 60 sq. ft. of boiler-heating surface.

Many types of grate bars have been designed according to varying ideas as to the best methods of burning different fuels. Stationary grates for hand-fired furnaces are made of cast-iron sections about 3 ft. long and several inches wide, as shown in Fig. 56. The bar *A* is called the common grate bar, while *B* is the herringbone type, and *C* is the pinhole grate bar which is designed for sawdust, wood refuse and the like. Lugs at each end of the bars rest upon bearer bars fastened to the walls of the setting. The length of the grate should not exceed 6 ft. with bituminous coal, 12 ft. for anthracite, otherwise the fireman cannot properly distribute the coal and break up the fire.

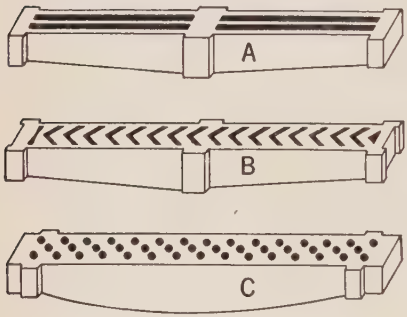


FIG. 56.—Types of grate bars.

In hand-fired plants, the use of sectional *shaking* and *dumping grates* presents a distinct advantage over stationary grates. The fireman can clean the fires without opening the furnace doors and without allowing the entrance of large quantities of excess air, which produce a waste of fuel and consequent low efficiency. For the boiler installation which is too small to warrant the use of a mechanical stoker, the shaking grate is often used. There is a large variety of shaking and dumping grates; in either type the grates are built up of a number of movable sections which are operated from the front of the furnace by one or more hand levers, as illustrated in Fig. 57. This design has been made as simple as possible by the elimination of small parts and complicated fastenings. The grate bars are of one-piece construction and are made with a lug at the lower end which is fastened to the shaking bar. The action of the grate is apparent from the illustration.

95. Mechanical Stokers.—A *mechanical stoker* is a power-operated fuel-feeding mechanism and grate. It is used for supplying solid fuel to a furnace, admitting air to the fuel for the purpose of combustion

and providing a means for the removal of refuse. To accomplish this result several distinct types of stokers have been developed to suit the conditions of various installations such as size of boiler served, fluctuations in load, draft available, quality of coal used, and space limitations. The first cost and the cost of operation and repairs of a mechanical stoker are considerably greater than for hand firing, but less labor and attendance are required for operation. A cheaper grade of fuel can be used, a higher efficiency attained, greater flexibility of operation assured, better furnace conditions can be maintained and less smoke formed than is usual with hand firing. Mechanical stokers are used in most power plants with boilers of 200 hp. or more. Furnaces for stoker-fired boilers should have a volume of 20 (underfeed) to 30 (chain grate) cu. ft. per pound of coal burned per minute. Where coal is used as fuel,

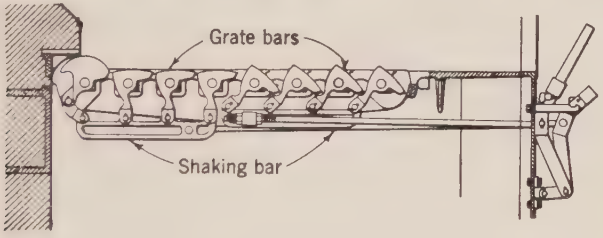


FIG. 57.—Erie City shaking grate.

the types of completely automatic mechanical stokers may be classified as chain grate, inclined frontfeed, inclined sidefeed and underfeed.

96. Chain-Grate Stokers. –In its elementary form, the chain-grate stoker consists of an endless grate composed of sections of bars passing over sprockets. The coal is fed from a hopper to the grate, Fig. 58, is ignited by passing under an ignition arch and is carried with the grate toward the rear of the furnace as the combustion progresses, gradually burning to ash. The ash and refuse are automatically dumped as the grate bars turn over the rear end of the stoker.

Stokers of this type have been widely used for burning bituminous coals of the Central West which are of the non-coking type and are high in volatile matter and ash. In recent practice they have also been used to advantage with anthracite screenings and coke breeze. Since the tars of a coking coal fuse at a temperature lower than that of volatilization, such coals have a tendency to clog the air passages of the grate. For natural-draft chain grates using coals of this kind, a coking plate is provided upon which the coal drops from the hopper. There it remains until the volatile matter has been driven off, after which the coked coal is delivered to the grate. Forced-draft chain grates burn coking coals

satisfactorily without agitation because the air supply can be regulated so as to consume the volatile matter where the fuel enters the furnace.

Chain-grate stokers are used with either natural or forced draft and are sometimes provided with draft compartments. These have individual damper control under the grate to give zone control of combustion, as shown in Fig. 58. At light loads this is an advantage, since it is possible to reduce the area on which the fuel is burned, thus giving a higher rate of combustion and usually higher efficiency. Chain-grate stokers are provided with an ignition arch to reflect the heat from the fire, down upon the fresh coal at the front of the grate. Furnace arches

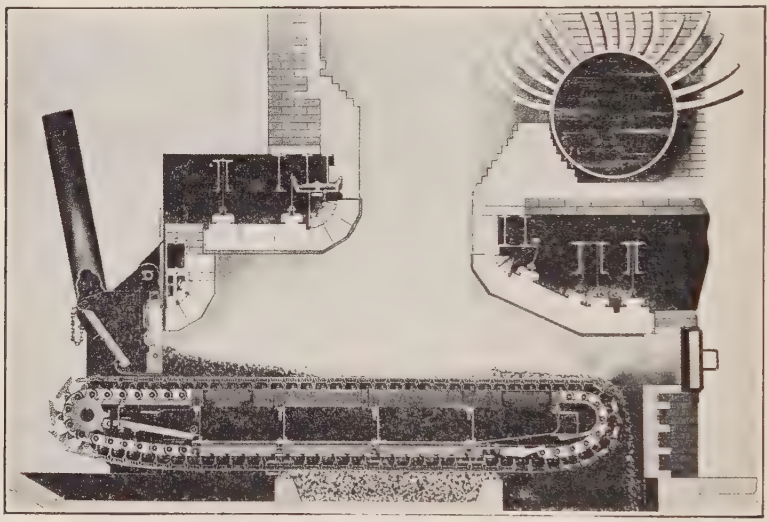


FIG. 58.—Harrington chain-grate stoker.

aid in maintaining the temperature for complete combustion before the gases reach the heating surface of the boiler. Incomplete combustion may occur with a chain grate when the fire is excessively forced. Extra air is likely to leak in if the fuel bed becomes too thin, thus causing a drop of temperature in the furnace and poor combustion results.

Any leakage of air around the back end of the grate is a source of loss, unless provision is made for heating this excess air and directing it into the path of the combustible gases. This leakage may be minimized by the use of a closed ashpit with a baffle placed just beyond the rear end of the stoker, a dump plate, or a water back placed on an overhung bridge wall, see Fig. 40. The water back, by presenting a water-cooled surface to which clinkers will not adhere, retains the incandescent fuel on the grate until it is more thoroughly burned and eliminates fusing of the

bridge wall. Modern chain-grate stoker installations are being provided with a rear arch in addition to the front arch, the rear arch deflects the excess air from the rear to the fuel bed and into the combustion space. This deflection also prevents burning of the grate links.

At the rear of the coal hopper is a cast-iron plate lined with firebrick that may be raised or lowered, thus regulating the depth of the fuel bed. With the forced-draft chain-grate stoker the rate of combustion varies from 40 to 60 lb. per sq. ft. of grate surface, per hour, with probably 45 lb. as the average. Using natural draft, the average combustion rate is lowered to about 30 lb.

97. Overfeed Stokers.—Stokers of the overfeed type are divided into two general classes depending on whether the coal is fed from the

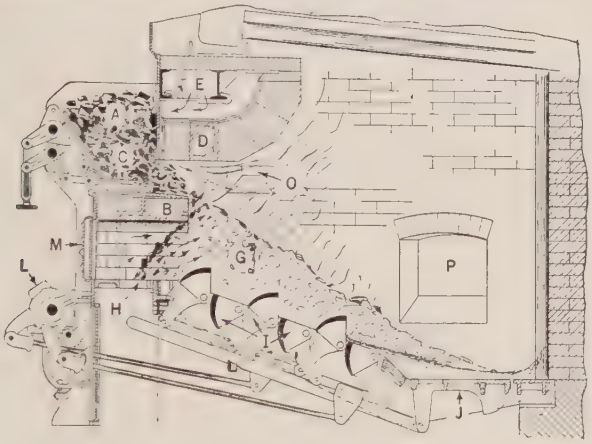


FIG. 59.—Marion overfeed stoker.

front or from the sides. Inclined *frontfeed* stokers receive coal from a hopper extending across the entire front of the furnace. In Fig. 59, coal is fed to hopper *A* by shovel or overhead conveyor and forced upon the arch jams *B* by mechanically-operated pushers *C*. The fuel is exposed to the radiant heat of the furnace and ignition arch *D*, and air for combustion enters from space *E*. When the coal reaches the fuel bed *G* ignition has taken place and a part of the volatile matter has been liberated. Air is supplied through *H* and by openings in and between grate bars *I* from the ashpit, additional air may be supplied by opening door *M*. The inclination of the grates is about 30 deg., which is sufficient, with the aid of the grate bar movement, to cause the fuel to descend. At the same time the working of the grate breaks up the coked fuel, providing air spaces through the fuel bed and sifting the ash to the space below. The coarser ash, clinker and some unburned fuel reach the dump plate *J*.

This stoker feeds the coal mechanically and stokes it mechanically, but dump plate J must be actuated manually for the removal of accumulated ash and refuse. Grate bars I may be operated by a hand lever inserted in the bracket L. Pusher plates C may also be actuated by hand in case of breakdown of the mechanical drive.

Fuel feed control is regulated by adjusting the stroke of the pushers for the grade of coal burned and by the motor speed for the load. The stoking control is such that the movement of the bars is comparatively fast so that they are not exposed in the hot fuel bed for more than a few seconds. The period between stoking is long enough to burn the coal added to the furnace during that time. The front of the furnace is made tight, so that forced draft may be used if desired. These stokers are adapted to low volatile coking bituminous coals, such as are found in the eastern states, which require agitation of the fuel bed and are easily ignited. When used with natural or induced draft, these stokers are not suitable to conditions where the load fluctuates rapidly. The maximum rate of combustion is about 40 lb. of coal per sq. ft. of grate surface, per hour, with forced draft and with natural draft about 20 lb.

Sidefeed stokers differ little in principle of operation from the frontfeed type. The Murphy stoker, Fig. 60, which is representative of this type, has coal hoppers H placed along the full length of each side of the furnace. This stoker uses two inclined grate surfaces so arranged as to form a V-shaped inclined fuel bed which is agitated by alternate movable grate bars G_m, the bars being parallel with the travel of the fuel bed. The alternate stationary grate bars G_s are ribbed to prevent the entrance of excessive air from below and, further, to reduce sifting of fine coal to the space below. The furnace is always of the extended or Dutch-oven type with an air-cooled arch A covering the entire space above the grates. The process of combustion is essentially the same as that in frontfeed stokers, but in this case clinker grinders C are usually employed to remove the ash automatically. The air used for cooling the arch is supplied (as preheated air) to the green coal through ducts in the arch plate. Due to the bottling effect of the furnace, the temperature runs higher than with the frontfeed type, making ignition more rapid but limiting the kinds of fuel used to those containing ash with a high temperature fusing point. Rates of combustion are practically the same for sidefeed as for the frontfeed stokers. The Murphy stoker uses natural or induced draft and is particularly suited for small or medium-sized boilers carrying a steady load.

98. Underfeed Stokers.—Underfeed stokers are designed on the principle that the hottest part of the fuel bed should be located where its radiant heat will impinge directly upon the boiler-heating surface. Coal

is pushed from the hopper into a retort by means of a reciprocating plunger. The retort is similar in shape to a trough, see Fig. 61, with the green coal forced in along the bottom. The fuel is gradually pushed up to the surface during the process of coking and ignition, and overflows at the sides or end to the ash dump plate or clinker grinders, as provided.

As the fuel bed carried on underfeed stokers is usually 24 in. or more in thickness, forced draft is essential for their operation, the air pressure below the fire being as high as 5 to 7 in. of water. With underfeed

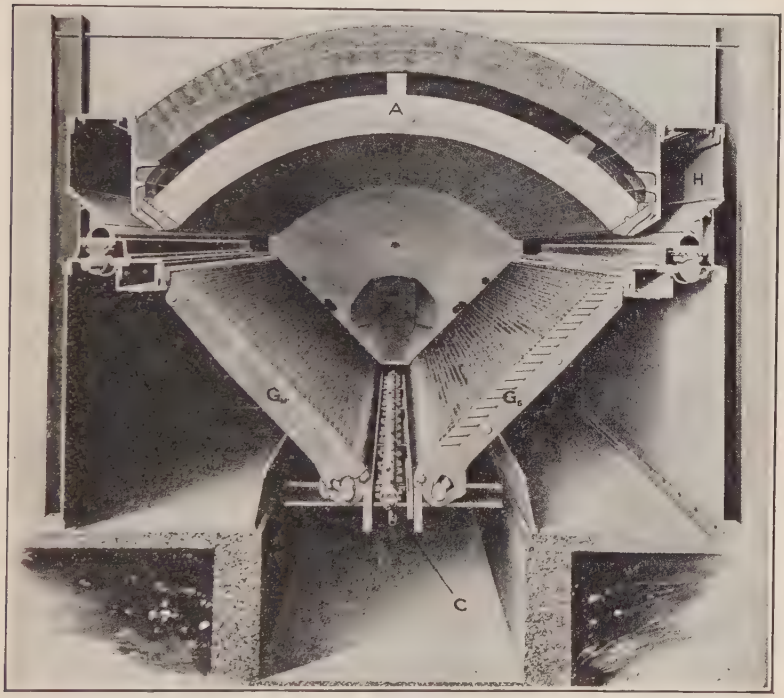


Fig. 60.—Murphy stoker and furnace.

stoker operation, there should always be a slight suction in the furnace. Air is forced into the fuel bed through tuyeres in the side of the retort. In passing through the fuel bed the air becomes heated before supplying oxygen for the combustion which is most intense at the upper surface of the fuel bed. The volatile gases driven off during the ascent of the fuel in the retort pass through this incandescent bed at the surface and due to the forced draft are thoroughly mixed with air so that combustion of the gases is complete at a short distance above the fuel bed. Furnaces for underfeed stokers are not as large as others of equal coal-burn-

ing capacity and they require no special arches to secure ignition and proper mixture of furnace gases.

There are two general classes of underfeed stokers, the *single retort* and the *multiple retort*. Fig. 61 shows a longitudinal section of a Jones "Standard" underfeed stoker, one of the earliest and simplest forms of the single-retort type. Coal is fed to the hopper and is moved forward and upward by the successive action of the steam-actuated ram and pushers A. Air for combustion is forced through tuyere openings in the hollow blast tiles located at a point above the fresh fuel in the retort, but below the fire. Thus, after the liberation of the volatile gases, the coke (fixed carbon) is burned at the top of the fire. The ash is forced to dead plates on the sides of the retort from which it is manually removed through doors. In a recent design the hand-cleaned dead plates have been replaced by a sloping combustion grate and a hand-operated dump plate. One or more stokers of this type may be installed in a furnace,

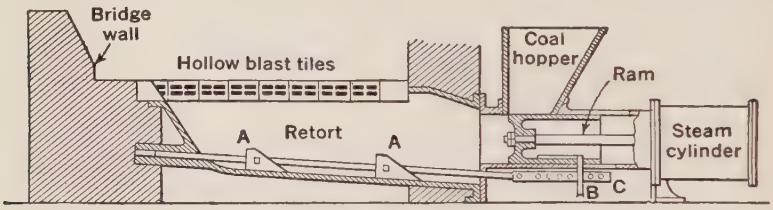


Fig. 61.—Jones "Standard" underfeed stoker.

depending upon the capacity of the boiler and width of the furnace. Dump plates for ash removal are located at the rear end when the retorts are used in multiple. The standard size of retort is 6 ft. long, 28 in. wide, and 18 in. deep. The supply of air and fuel may be automatically controlled by the steam pressure in the boiler. The result is that the stoker, if correctly installed and operated, effects complete and smokeless combustion. The Jones "Standard" underfeed stoker is applied to boilers of 50 to 150 hp. It has a fuel-burning capacity of 800 to 1200 lb. of coal per hour.

The Jones "Lateral Retort" is a multiple-retort underfeed stoker. Fig. 62 shows a multiple-retort unit consisting of one main retort extending from the front wall of the furnace to the bridge wall, with four lateral retorts at right angles to the main retort. Coal is fed into the main retort from a fuel hopper by a ram, similar to the feeding mechanism shown in Fig. 61. The actuated pusher blocks *A* in the main retort assist the coal to move lengthwise. The coal is then forced sidewise into lateral retorts by means of lateral pusher blocks *L*. These are operated by an auxiliary steam cylinder located outside the boiler front.

Hand-operated side dump plates P for ash removal are located at the end of the lateral retorts. A transverse section of a double lateral

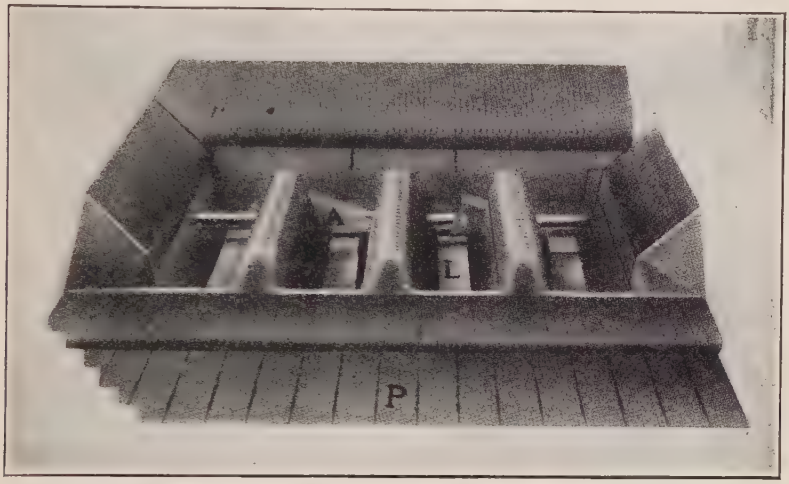


Fig. 62.—Jones "Lateral Retort" underfeed stoker.

retort stoker is shown in Fig. 63, in which two sets of lateral retorts branch from the main retort. The approximate movement of the lateral pusher blocks L is indicated by the white dotted line in Fig. 63. The

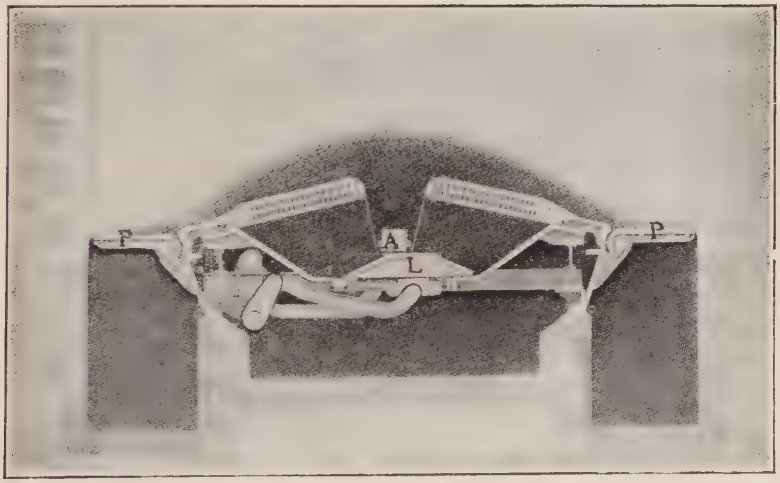


Fig. 63.—Transverse section of Jones "Lateral Retort" stoker.

pusher blocks L are operated alternately in the lateral retort, Fig. 62. This slicing action keeps the fuel bed well broken up, allowing the air to

thoroughly mix with the coal. The lateral retort underfeed stoker is recommended for use with boilers ranging from 150 hp. to 500 hp.

The Taylor stoker, as illustrated in Figs. 64 and 65, is typical of the modern inclined multiple-retort underfeed stoker. It consists of alternate retorts and tuyere boxes. The entire bottom of each retort is formed by a series of overlapping and interlocking (adjustable) distributing rams, driven by the main feeding ram. The volume of each retort can be varied by changing the position of the distributing rams. The coal is forced into each retort by a reciprocating ram or piston. When the ram has moved about two-thirds of its stroke (this distance is adjustable) it

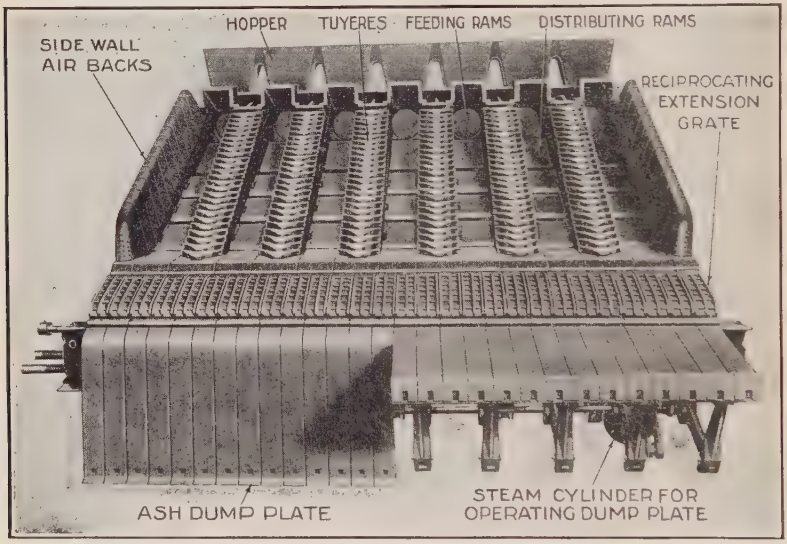


Fig. 64.—Taylor underfeed stoker.

starts to drive the distributing rams; during the remainder of the stroke the main and distributing rams move together, carrying the entire fuel bed with them. Because of the sloping faces of the distributing rams the fuel bed is raised upward as it is pushed toward the end of the retorts. The movement is small and gradual, and it has a mild slicing action which aids combustion. The fuel bed being tightly packed, remains in place when the distributing rams are withdrawn on their outward stroke, leaving a small space along the entire under side. Green coal is forced down into this space during the first part of each inward stroke of the feeding ram, before the distributing rams start to move. Step by step, the coal is pushed up into the coking zone where it is gradually coked by the heat of the fire above. The volatile gases released pass upward through the burning fuel. Then the fuel is forced up into the zone of

combustion where it forms the bed of incandescent coke and works slowly downward, toward the ashpit, aided by the action of the distributing mechanism and gravity. To burn out any carbon remaining in the ash after it passes out beyond the retort section, a movable overfeed extension grate is provided, Fig. 64. This grate is connected to the bottom distributing ram as shown in Fig. 65, and in this way is given a reciprocating motion that coordinates with the distributing system in moving the lower portion of the fuel bed. Ash disposal is accomplished either by a hand- or steam-operated dump plate or by the Taylor rotary ash discharge (toothed crusher rolls). The dump plate is arranged to swing above the horizontal to clean the bridge wall of clinker adhesions. Where loads are nearly uniform, the rotary ash discharge is applicable,

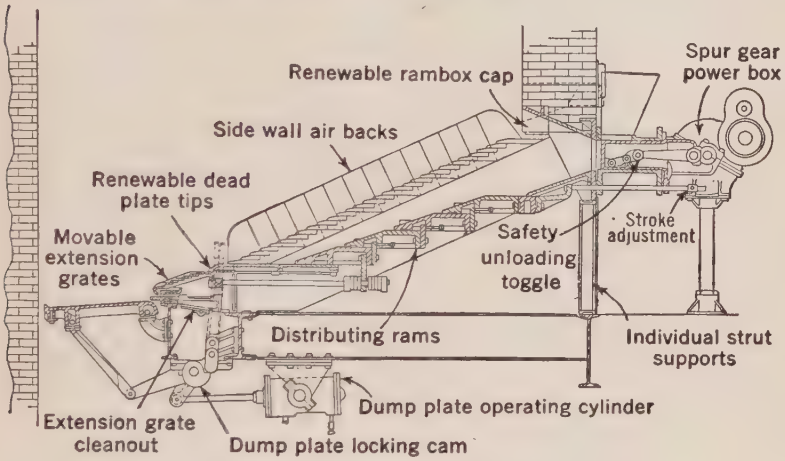


Fig. 65.—Side elevation of Taylor underfeed stoker.

but where extreme flexibility is desired the dump plate is preferable. The area under the entire fuel bed supporting structure is enclosed, air under pressure is supplied to this space and is introduced at the rear. This gives the maximum cooling effect on that part of the grate which is subjected to the greatest heat and preheats the air before it passes through the tuyere box openings into the furnace. Air is also supplied to the fuel through openings in the distributing blocks.

Underfeed stokers are particularly well adapted for coals high in volatile constituents which have caking characteristics and the ash content of which is low and non-clinkering. The continual agitation of the fuel bed keeps it porous even with coals that cake. Where quick steaming is desired to meet high overloads, the underfeed stoker may be used to advantage. Eastern coals may be burned continuously at a

rate of 60 lb. per sq. ft. of grate surface, per hour, while for short periods of time the combustion rate may be increased to 70 lb. With western coals, the combustion rate is lowered to around 40 lb. Fires on underfeed stokers are easily banked and may be brought into service again in a short time.

99. Pulverized Fuel Burning.—Boiler plants using pulverized coal (also called powdered coal) as a fuel require apparatus to convert the raw coal into a powder and convey it to the furnace, into which it is delivered as required and burned in suspension. The systems used to prepare and burn pulverized fuel may be divided into two classes:

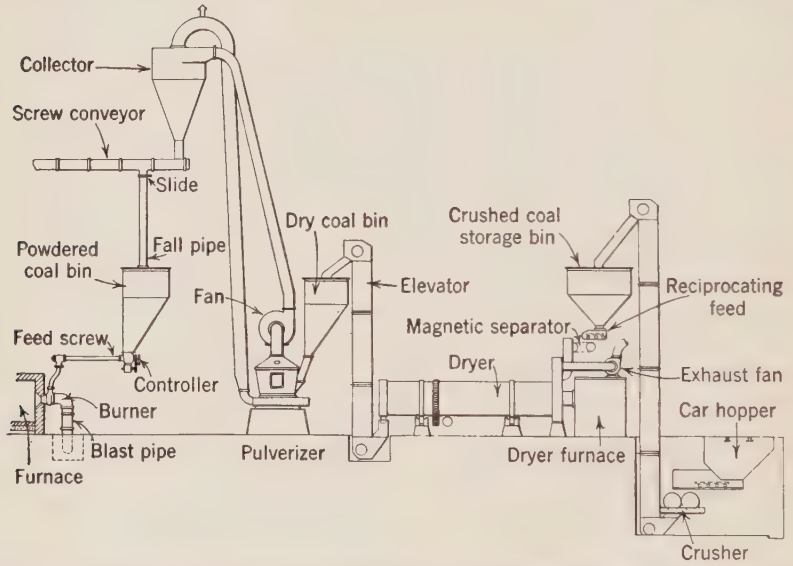


FIG. 66.—General arrangement of equipment for bin and feeder system of preparing and burning pulverized coal.

(1) The unit system, in which one machine (pulverizer) prepares and delivers the coal to the furnace with the necessary air for combustion. (2) The multiple (bin and feeder) system, with its independent preparation plant, pulverized coal transport system, pulverized coal bins and feeders. The multiple system is the more common of the two systems; however, in recent years the unit system, on account of its simplicity and low cost, is being installed more frequently.

With the bin and feeder system, shown diagrammatically in Fig. 66, the coal is first crushed to a general size of $\frac{3}{4}$-in. to 1-in. lumps. It is next passed over a magnetic separator to remove tramp iron, such as bolts, nuts, rivets, etc. If this iron were not removed, damage would result

to the pulverizer. After leaving the magnetic separator the coal passes through the rotary dryer, the latter being separately fired as shown, or it may be heated by waste flue gases or steam. The dry coal enters the pulverizer and when sufficiently ground, 65 per cent passing through a 200-mesh screen, the coal is lifted by a current of air through a fan to

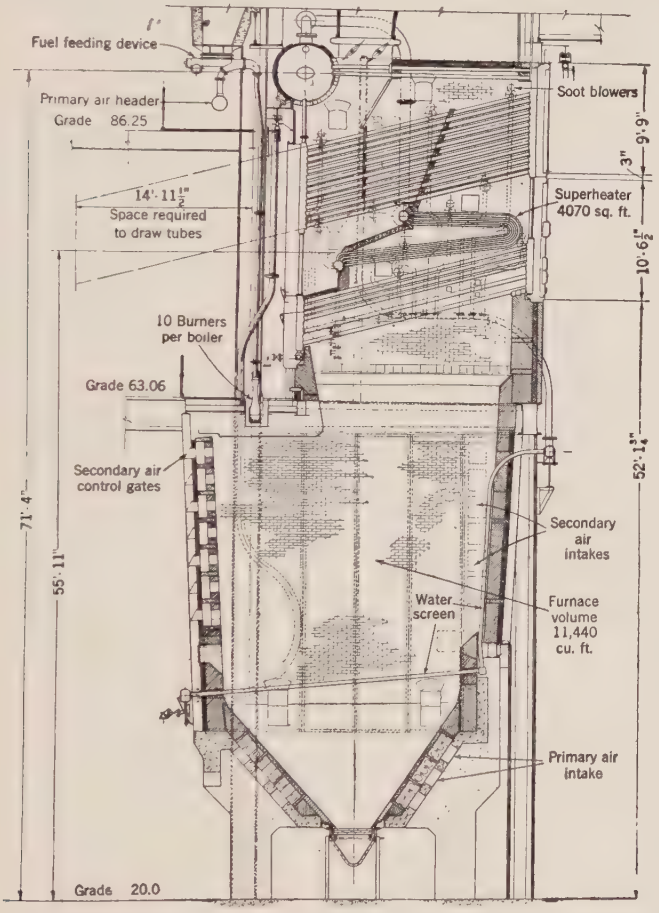


FIG. 67.—Pulverized-coal furnace, Combustion Engineering Corporation.

the vented collector where the air and coal separate. From the collector the coal falls down to the screw conveyor, the latter distributes the coal from the central pulverizing plant to the powdered coal bins situated near the furnaces. The coal is fed to the burners by a feed screw equipped with a variable speed control. When the powdered coal reaches the burner it meets a current of air, the coal and air mixing as

they enter the furnace. A secondary supply of air is usually provided through annular ports around the burner and sometimes through openings in the furnace walls as shown in Fig. 67.

In the unit system, Fig. 68, the essential elements are reduced to a minimum. The unit pulverizer mill readily adapts itself to the use of preheated air, thus eliminating the necessity of drying the coal. No

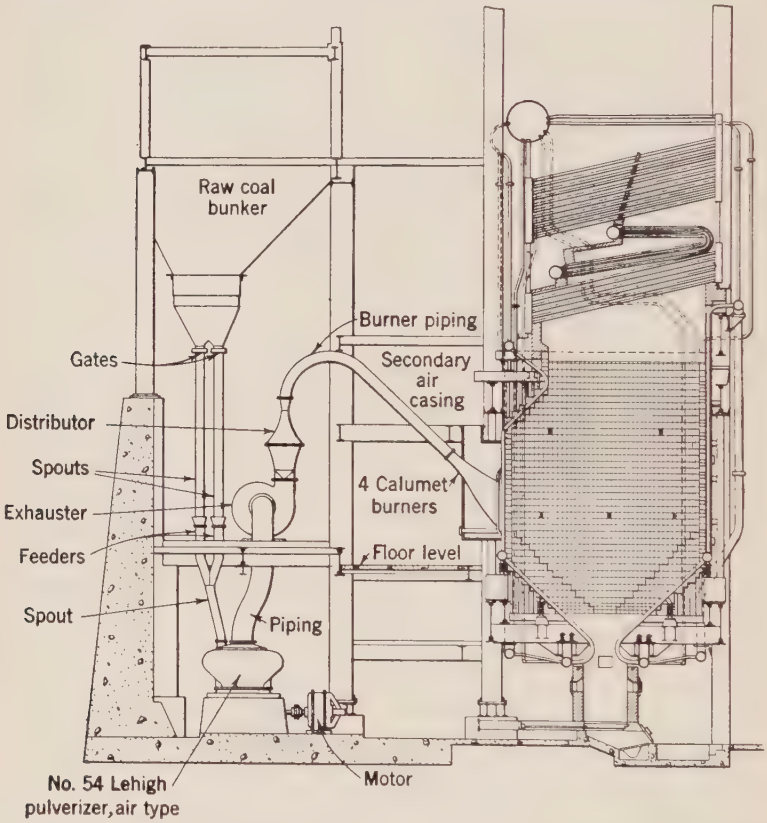


Fig. 68.—Central station installation, unit system of pulverized-coal equipment.

pulverized coal collectors are necessary. The system requires the following important parts: raw coal conveying system and bunker, air-swept pulverizer, exhauster fan, secondary air fan, and burner.

A pulverized coal furnace should be so designed that any kind of coal available may be burned efficiently and at the desired rates of combustion, without smoke and undue maintenance cost. The volume of the furnace depends primarily upon the fineness of the fuel particles and the method of introducing air and fuel into the furnace. These

are coordinated with the type of wall construction and the rate of heat liberation; 50 to 70 cu. ft. of furnace volume per lb. of coal burned per minute are average values used. The ordinary refractory walls wash away when using a coal having a low temperature fusing ash. The temperature of the lining may be maintained below that of the slag by placing water screen tubes, Fig. 67, or a radiant heat superheater, *R*, Fig. 72, in front of the brick walls. Recessed water tube refractory faced walls and air-cooled walls are also being used. The successful collection and disposition of the ash from burning pulverized coal is another problem. Some of the earlier plants used a water screen near the bottom of the furnace. In falling through this screen, the ash particles were cooled and collected in the ashpit as dust. However, at heavy loads the flame would penetrate the screen and slagging on the tubes and in the ash pit generally resulted. A recent development is the use of cast-iron-faced water-cooled ashpits built in the form of a V, Fig. 68. Thus the whole furnace volume is available for combustion and slagging conditions are minimized. Although removal of the ash in granular form is customary, attention is being directed to horizontal firing and the desirability of tapping the ash off in the form of molten slag, as is done in some metallurgical processes.

Advantages claimed for the use of pulverized coal include: complete combustion, absence of smoke, cheaper fuels can be used, easy control of fuel and air supply, no standby losses, low labor cost, and usually higher efficiency. Some of the disadvantages are: high first cost, cost of fuel preparation, storage limitations of pulverized coal because of caking or fire hazard, furnace depreciation, disposal of ash and slag, and the possibility of refuse discharging through the chimney. The elimination of ash discharge into the atmosphere has in some instances become a decided civic problem and legislation has been passed regulating its control. Many comprehensive efforts have been made to collect this ash by the use of electrical precipitation, water spraying and mechanical separation in baffle and cyclone collectors.

100. Coal and Ash Handling.—The handling of coal and ashes in the small plant, less than 500 hp. is frequently accomplished with the aid of shovel and wheelbarrow. Larger plants are generally equipped with conveying machinery to insure continuity of operation and to dispense with unreliable labor. The method of delivering the fuel to the furnace and of conveying the ashes away should be one that will give good results for minimum cost. Some of the methods used are: (*a*) wheelbarrow and hand shoveling, (*b*) continuous conveyors, endless belt and pivoted buckets, (*c*) pneumatic as used for pulverized coal, (*d*) hoists and cranes, and (*e*) hydraulic sluice. Ashes are sometimes handled by the same

conveying system used for the coal but in many plants the disposal of each is taken care of separately because of greater flexibility of operation, and on account of the abrasive and corrosive action of the ashes. Fig. 69 shows a diagrammatic arrangement of a pivoted-bucket conveyor which is one of the popular methods for handling coal from car to furnace. The buckets are hung on pivots and suspended between two endless chains and maintain their upright position at all times. The chain is driven by a motor and sprocket wheels. The point of delivery of the coal is controlled by the location of the movable discharger.

101. Air Preheaters.—As used with a boiler, the purpose of an air preheater is to extract heat from the flue gases and transfer it to the air used for combustion of the fuel. The greatest source of loss in connec-

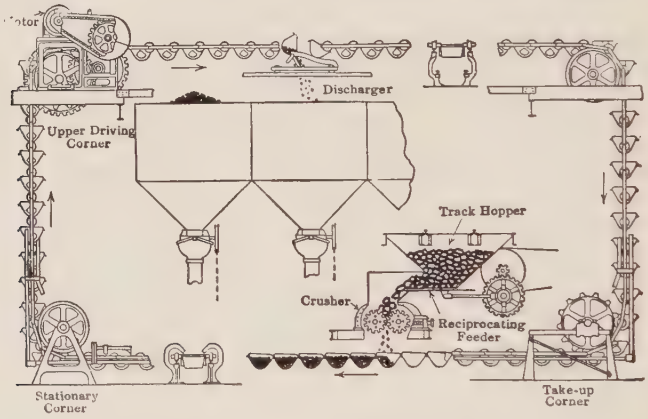


Fig. 69.—Peck pivoted-bucket conveyor.

tion with steam boiler operation is the quantity of heat carried away by the flue gases. The forcing of boilers to high overloads and the desirability of complete combustion with a minimum air supply results in high exit gas temperatures. Economizers, described in Chapter IX, have been installed in most plants of any importance in order to utilize some of the heat in the exit flue gases. The use of exhaust steam or steam from a bleeder turbine for heating the feedwater may eliminate to some extent the necessity of using an economizer and as an alternative a preheater may be used. Advantages derived from the use of preheated air for combustion are: heat conservation, improved combustion, successful burning of low grade fuel, increased efficiency and probable increased capacity. The rate of ignition and combustion is greatly accelerated by the use of preheated air, making possible higher combus-

tion rates per unit of furnace volume and, therefore, smaller furnaces. The hotter air supply increases the furnace temperature and hence increases the radiant heat transmission to the boiler.

Air preheaters use either the convection or regeneration principle of heat transmission. The *convection* type may be subdivided as plate or tubular, whereas the regenerative type employs a metal rotor which is alternately heated by the gases and then cooled by the air. Fig. 70 shows a side elevation of a tubular type air preheater using the counterflow principle of heat transmission. The hot gases enter the lead-coated tubes at the bottom and leave at the top of the heater as indicated. The cold air enters at the upper left side, flows around the outside of the tubes and leaves at the lower left. Baffles are supplied as shown in Fig. 70 for the purpose of directing the flow of air so that the maximum heat transfer is obtained; the baffles are arranged to keep the velocity of the air as nearly constant as possible. In the plate type of heater the gas and air travel through alternate passages between the plates.

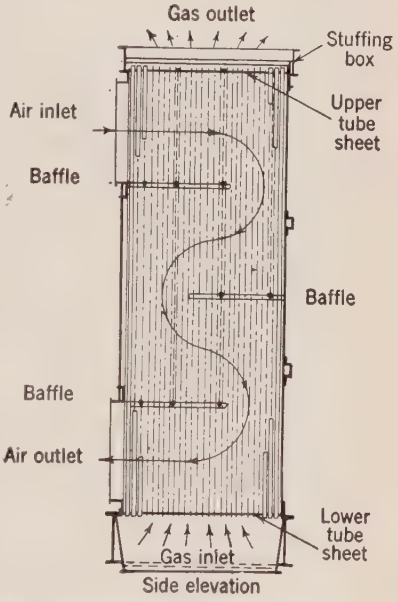


Fig. 70.—Sturtevant tubular air preheater.

The *regenerative* type of preheater is illustrated in Fig. 71. Fresh air is drawn by a fan into the upper portion of the casing which is divided into two parts by a vertical partition. The air is confined to one side of the partition and passes downward through a porous cylindrical drum or rotor to a semicircular chamber in the lower part of the housing. The flue gases from the boiler traverse the apparatus in a reverse path. They enter the lower semicircular chamber as indicated and pass upward through the drum rotor to the upper chamber, whence they are exhausted and impelled to the chimney by a second fan. The drum is kept in continuous slow rotation so that the part heated by the flue gases is constantly passing to the other side of the device and gives up its heat to the cold air sweeping through it. Similarly, the cool part of the rotor is returning to be reheated by the flue gases. Cleaning and corrosion are important factors affecting the performance of air heaters, especially the convection type.

102. Superheaters.—Superheated steam has a temperature higher than that of saturated steam at a corresponding pressure. Superheated steam greatly improves the economy of reciprocating steam engines and turbines, and is extensively used in large modern power plants. The increased economy may be effected in general by increased overall boiler efficiency, reduced condensation losses in steam pipe lines, reducing

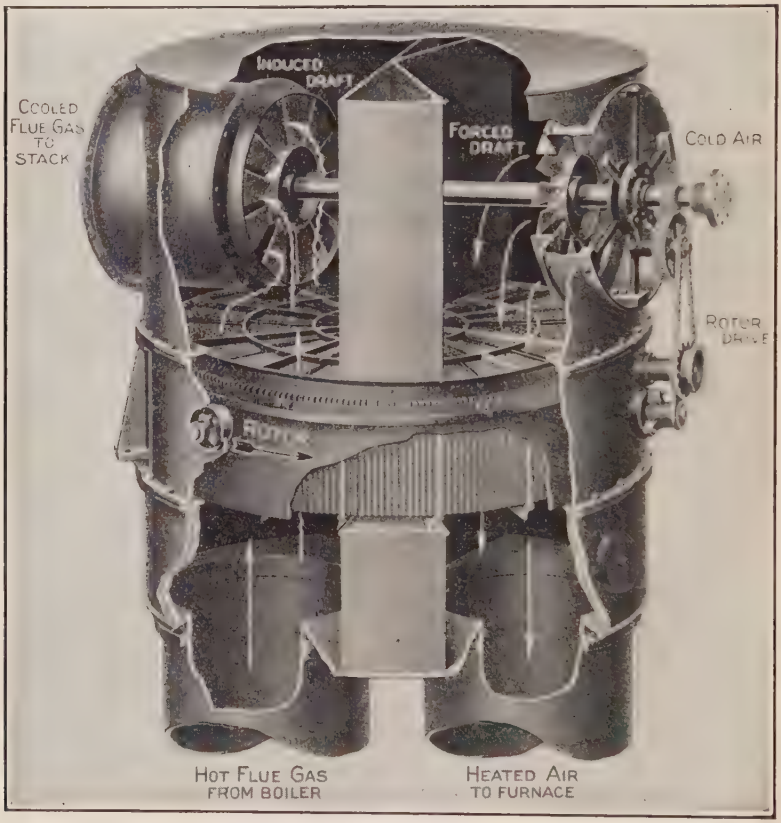


FIG. 71.—Ljungström air preheater.

steam consumption of the prime movers and by increasing the capacity of the plant. The purpose of a *superheater* is to impart additional heat to steam after it has left the boiler in which it was generated, thus increasing the steam temperature. The essential requirements of a successful superheater should include safety in operation, accessibility, protection against overheating, improved efficiency of the combined boiler and superheater, regulation, provision for keeping units clean and allowance for expansion and contraction of all parts.

Superheaters may be classified according to the manner in which the heat is transferred to the steam, as *convection* and *radiant* types. Convection superheaters may be independently fired or integrally built within the boiler setting. The independently-fired superheater is not widely used, being confined in general to special and limited cases where a small amount of superheated steam is desired at a high temperature. The advantages of regulation and making repairs without affecting boiler operation are offset by higher fuel cost, lower economy, separate attention, and additional floor space required. Integrally built convection type superheaters receive heat from the same flue gases and same furnace as the boiler. They may be located among the banks of the boiler tubes, Figs. 36 and 40, or at the top of the tubes and below the drum as *C* in Fig. 72. The radiant superheater, *R*, in Fig. 72, is placed on the furnace walls, and absorbs heat by radiation from the fire. With convection superheaters the temperature of the superheated steam increases with an increase in the boiler capacity, see Fig. 73. Radiant superheaters have the opposite characteristic, i.e., the superheat decreases as the boiler output increases. Practically constant superheat is obtained at all loads, Fig. 73, by a combination superheater, part of the surface being a convection superheater connected in series with the other smaller part installed as a radiant superheater. A combination superheater is shown in Fig. 72.

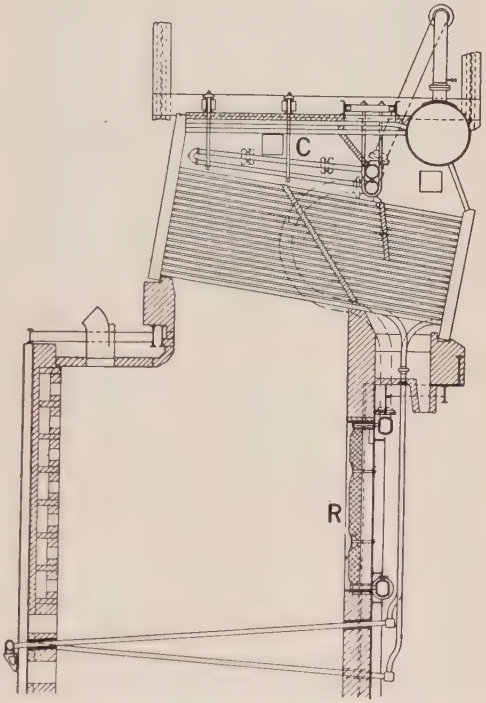


FIG. 72.—Convection superheater and radiant superheater.

Fig. 67 illustrates the application of an *interdeck* convection superheater to a cross-drum Babcock and Wilcox boiler. The steam collected in the dry pipe of the boiler drum passes downward to the upper manifold of the superheater. From there the flow is through the U-shaped steel tubes into the lower manifold, thence to the superheated steam header

serving the turbine or engine. For smaller boilers the convection superheater may be located above the boiler tubes between the first and second passes as shown at C, Fig. 72. However, more uniform superheat at various boiler outputs is obtained by placing the superheater between the tubes, Figs. 36, 40 and 67.

A radiant heat superheater is shown at R in Fig. 72 installed in a pulverized coal boiler setting. This superheater is placed to form a part of one of the walls of the furnace for the absorption of heat by direct radiation from the fire. The inlet steam header is at the top and the outlet superheater header is located at the bottom. Radiant superheaters serve as a protection for the furnace wall to which they are fastened and high superheats are usually obtained. Radiation to the

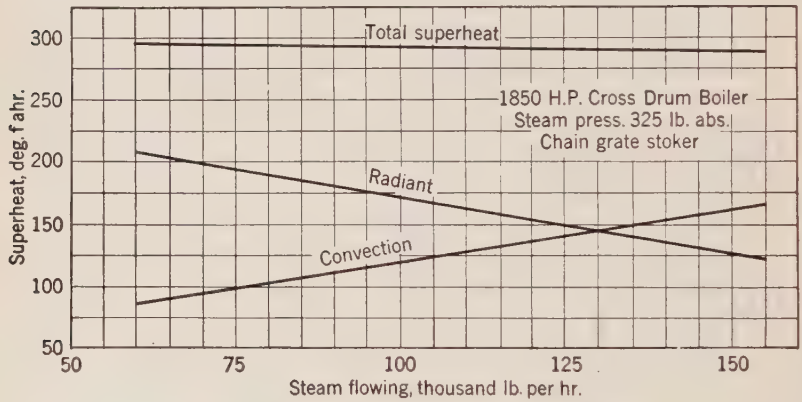


Fig. 73.—Typical curves for combination superheater.

superheater depends on: (1) temperature, position and radiating properties of the flame, (2) combustion rate, (3) excess air furnished, and (4) heating surface of the superheater.

103. Steam Separators and Traps.—Saturated steam flowing from the boiler to engines or turbines may contain water because of priming in the boiler or condensation in the pipe lines. With the small clearances now allowed in reciprocating engines, water in steam may break either the valve, cylinder head or piston; in smaller but continuous quantities it may cause erosion of valves, valve seats and cylinders. The use of wet steam also requires more cylinder oil and there is a constant loss in economy from increased initial condensation. With steam turbines, the moisture content reduces efficiency and causes erosion of the blades. The purpose of a live *steam separator* is to remove the water from saturated steam by the principle of reversed flow, baffle plates, centrifugal force, or by mesh separation. Figs. 74 and 75 show sections through typical

steam separators of the baffle type. The moisture is thrown out by the reversal of the steam flow or adheres to the surfaces and collects in the water chamber at the bottom of the separator, from which it may be removed by a steam trap. A *steam trap* is an automatically controlled valve, the function of which is to allow the discharge of separated water and to prevent the escape of steam. Steam separators, steam radiators or heating coils, steam-jacketed kettles or any steam apparatus from which it is desired to remove the condensate as rapidly as it accumulates should be drained by the use of a steam trap. Traps may be classified according to the principle of operation as float, bucket, bowl and expansion. Fig. 76 shows a section through a typical float trap in which water gravitating through connection *I* accumulates in the trap. The movement of float *C* opens the valve *F*; thus the water is discharged through *H* by the steam pressure acting upon the surface of the liquid. Water is removed until the level in the trap lowers the float enough to close valve *F*. Part *A* is used for manually opening valve *F*, *B* is an air vent and *J* a sediment discharge connection.

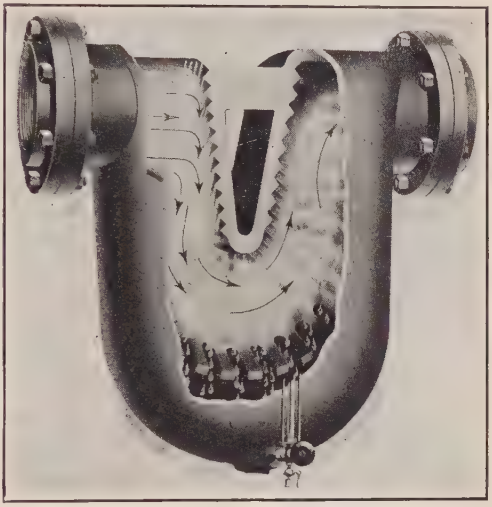


Fig. 74.—Wright–Austin horizontal steam separator.

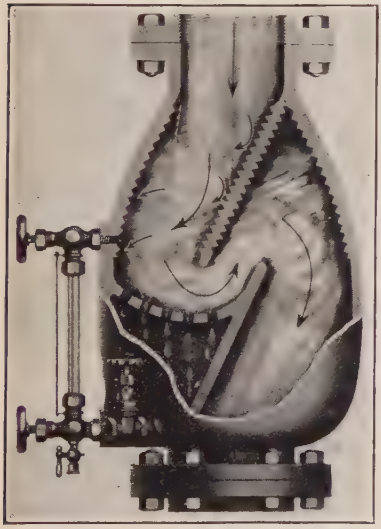


Fig. 75.—Wright–Austin vertical steam separator.

A typical bucket trap is shown in section by Fig. 77. The water of condensation enters the body of the trap and fills the space around the bucket. The bucket then floats and forces the valve against its seat, as shown. When the water rises above the edges of the bucket, an over-

flow into the bucket causes it to sink, thereby withdrawing the valve from the seat. This permits the steam pressure acting on the water surface to force the water through the valve opening to the outlet connections. When the bucket is empty, it rises and closes the valve and the cycle is repeated.

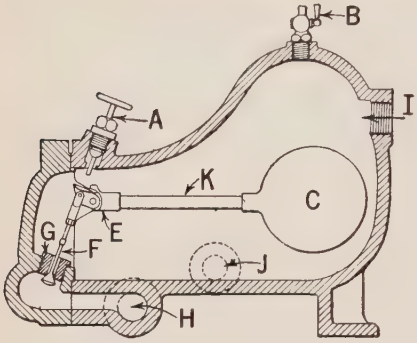


FIG. 76.—McDaniel float trap.

Fig. 78 shows a section through an expansion or thermostatic trap. The Sarco trap consists of a steam pipe body in which is inserted a cartridge containing a heavy hydrocarbon oil and a spirally corrugated tube. One end of the cartridge carries the valve head. When water collects and cools the oil, the oil and tube contract and the valve opens. The condensation is ejected by the pressure behind it and when steam comes in contact with the cartridge there is expansion and the trap closes.

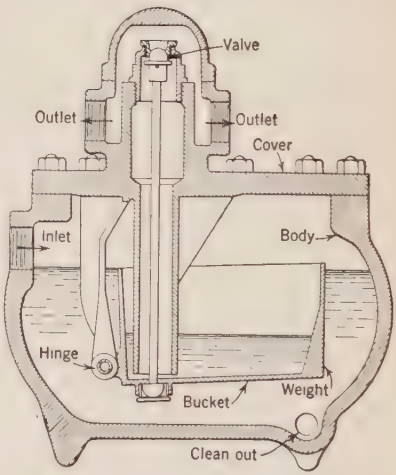


FIG. 77.—Squires bucket trap.

104. Steam Purifiers.—All boilers, regardless of design, may deliver steam containing moisture, especially if fed with a water that foams upon boiling. The percentage of moisture so delivered will depend primarily upon the rating being developed, the water level in the boiler and the amount of impurities in the boiler water. The *steam purifier* is a separator installed inside the boiler drum in the same relative position as the dry pipe and so constructed that moisture and solids are removed as the steam passes through the purifier. The removal of impurities is desirable because of the resultant difficulties that may arise from deposits in the

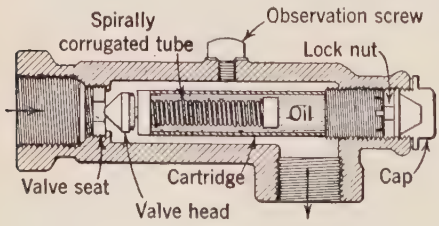


FIG. 78.—Sarco expansion trap.

pipe lines, traps, superheaters, valves and prime movers. The maintenance of clean superheating surfaces permits better heat transfer and consequent higher superheat. The use of clean dry steam in a turbine aids in the maintenance of the correct contour of passages by reducing erosion and corrosion, and prevents the depositing of foreign materials on the nozzles and blades.

The Hagan steam purifier, Fig. 79, removes moisture and suspended solids from the steam by centrifugal separation. The steam to be cleaned enters the purifier through the directing vanes, impinges against the walls of the separating chamber and is given a whirling descending motion as indicated by the arrows. The moisture is thrown against the wall and is directed and gravitated toward the tangential discharge outlet, to a settling basin or blow-off. The steam reverses its direction and ascends as a revolving core surrounded by the revolving annular body of steam moving in the opposite direction. Clean dry steam leaves the purifier at the top and is delivered to the steam header or superheater. " Tracyfier " is the name given to a steam purifier which consists of a number of rows of baffle plates so spaced that the steam is broken up into small thin ribbons about one-twentieth of an inch in thickness. The moisture adheres to the baffles. The velocity through the channels is low so that the moisture is not picked up again by the steam but the moisture and solids gravitate to the bottom of the chamber and discharge to waste.

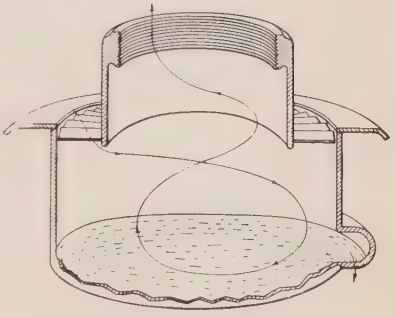


FIG. 79.—Hagan live steam purifier.

105. Steam Injectors.—An *injector*, Fig. 80, is a device consisting of a series of nozzles so arranged that a jet of steam expanding therein attains great velocity. The rapidly flowing steam meets a mass of water and condenses, and imparts sufficient momentum to the water to force it into the boiler. A simple form of steam injector is shown diagrammatically in Fig. 80. The steam flowing through nozzle A loses pressure and gains velocity as it enters the combining tube C. A reduced pressure is created in B, thus causing the water to rise until it comes in contact with the steam at the entrance to tube C. The steam at a high velocity condenses on meeting the water in C and imparts momentum to it. The delivery tube D receives the combined jet of water and condensed steam and converts the velocity of the jet to a pressure high

enough to force the water into the boiler. When starting, steam and air are forced to overflow into chamber E and through overflow valve V

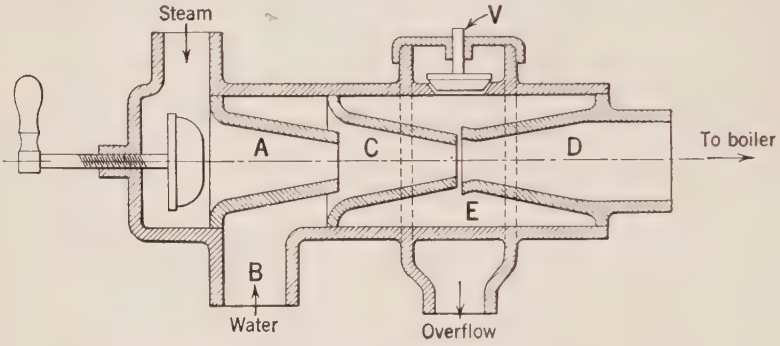


FIG. 80.—The steam injector.

to the atmosphere until water comes in contact with the steam, then the injector discharge pressure increases and the water is forced into the boiler. A partial vacuum is formed in space E and valve V closes after the flow has been established.

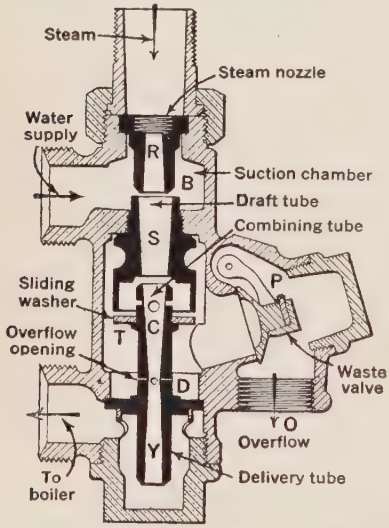


FIG. 81.—Penberthy single-tube automatic injector.

Injectors may be classified as *lifting* and *non-lifting*, depending on whether or not a partial vacuum is created in the suction pipe when starting. A non-lifting injector must always be placed below its source of feedwater supply because of its inability to create a partial vacuum. Injectors that have one set of nozzles for lifting the water and another set for forcing it into the boiler are called double-tube injectors. Those which accomplish the same result with one set of nozzles are called single-tube injectors, Fig. 81. The injector is an efficient and convenient device, cheap and compact, with no moving parts, and delivers hot water to the boiler. The use of injectors on steam locomotives is almost universal, but in stationary practice their usage is limited to small boilers or as a reserve with feed pumps. Some of the disadvantages of an injector are: inability to handle hot water, difficulty of maintaining a

continuous flow under extreme load variation, and the unreliability of operation under certain conditions. From a purely thermodynamic standpoint, the efficiency of an injector is nearly 100 per cent, since the heat of the steam is returned to the boiler, less a small radiation loss. However, as a pump the injector requires more steam per pound of water handled than is required by inefficient plunger pumps.

CHAPTER IX

FEEDWATER TREATMENT AND FEEDWATER HEATERS

106. Feedwater Impurities.—Chemically pure water is the ideal boiler feed supply. In commerical power plants such water is not available and waters of varying degrees of purity must be used. The need for water of greater purity is more urgent in large boiler units and in plants where boilers are driven at high steaming rates. Small and moderate sized plants, operating without undue forcing and with intermittent service, may be able to use untreated water of good quality or poor water with a limited amount of treatment.

All water in nature is more or less contaminated. Falling rain drops absorb oxygen, carbon dioxide, nitrogen, dirt, and other impurities from the air. Water thus polluted is a solvent to a greater or less extent of the minerals of the earth. This condition is augmented by the addition to the water of acids formed by decaying organic materials. Furthermore, the waters of lakes and rivers are frequently contaminated with industrial wastes and sewage so that they carry acids, alkalies, oil, dirt, organic matter, and debris, in addition to the impurities collected due to natural causes.

Feedwater impurities may be responsible for the following ill effects in boiler and power-plant operation:

(1) Reduction of the amount of heat transmitted to the boiler, due to scale formation on the boiler heating surfaces.
(2) Tube and plate failures caused by the reduced amount of heat transmitted through them.
(3) Embrittlement or corrosion of the boiler steel.
(4) Bad operation, foaming and priming.
(5) Cost of cleaning, repairs, inspection, maintenance, and reserve equipment.
(6) Heat losses due to frequent boiler blow-downs.
(7) Loss of economy of steam using equipment caused by dirty steam.
(8) Additional cost of condensing equipment for gas-laden steam.

Most of the foregoing ills may be attributed to one or more of these items, scale, corrosion, embrittlement, or foaming and priming.

Scale is a deposit, on the internal surfaces of the boiler, of mineral materials and dirt. The nature of the scale varies from soft to very hard and flinty, and its effect, even with small deposits, is to insulate the metal surfaces and to reduce the boiler capacity. Such insulation may cause overheating and weakening of the boiler tubes and shell plates.

Corrosion produces a wasting away of the boiler parts with a reduction of their strength. Corroded boilers in some cases may be operated with reduced pressures depending upon the extent of the decaying action, in other cases corrosion may necessitate the condemnation of the boiler for steam-generating purposes.

Caustic embrittlement occurs in boilers operating with waters abnormally high in free sodium bicarbonates or waters carrying free sodium carbonate without sulphates. These materials hydrolyze under heat and pressure into sodium hydroxide and liberate free carbon dioxide. Steel is likely to be affected by the hydroxide at overstressed localities and be rendered brittle and unreliable under pressure.

Foaming and priming may occur in combination. Suspended organic matter, some oils in the presence of soda salts, or waters overtreated with soda ash may cause the water in the boiler to fill the steam spaces with a mass of frothy bubbles due to the surface tension of the water at the steam disengaging area. When this occurs the term foaming is applied. Priming is the discharge of slugs of water with the steam when foaming occurs, or it may occur in boilers having small steam and water disengaging surfaces. The rapid ebullition of water in such boilers causes water to be thrown up into the steam space and to be carried away with the steam. Both foaming and priming cause dirt and other objectionable material to be carried by the steam to foul superheaters, traps, steam lines, and turbine blading.

107. General Effects of Impurities.—Some of the general effects of the boiler feedwater impurities listed are enumerated hereafter. *Oxygen* induces pitting and corrosion of all metal parts with which it comes in contact. This is especially true at high temperatures. *Carbon dioxide* alone is likely to produce corrosion or to enter into combination with other materials to produce scale-forming compounds. *Gases* such as oxygen, nitrogen, carbon dioxide, and others not condensable at the usual condenser pressures and temperatures, which may be carried by the steam, impose an additional load upon the air ejectors or air pumps of condensers. *Dirt* and *mud*, unless washed out of boilers frequently, are likely to bake on the interior surfaces and produce scale. *Acids* pro-

duce corrosive action within the boiler, which results in the destruction of the boiler parts or else renders them unsafe for operation. *Alkalies* induce bad boiler operating conditions such as foaming and priming, and also under certain conditions may produce either corrosion or embrittlement of the boiler parts. *Oils* produce foaming and priming in a boiler and may coat the interior surfaces so that the heat transfer from the plates and tube walls to the boiler water is greatly retarded. Burning and weakening of the boiler parts results. *Fatty oils* break up into acid compounds producing corrosion. *Dissolved organic matter* may break up into acids or into alkalies and thus produce very undesirable results. *Dissolved mineral materials* are usually precipitated by heat or left as a residue after the evaporation of the water and thus produce boiler scales of different degrees of hardness.

108. Water Hardness.—Hardness is a relative term applied to waters. One method of indicating the degree of water hardness is by the use of a standard soap solution. The indication is obtained by noting the quantity of the soap solution necessary to produce a lather in 100 c.c. of the water. Hardness is indicative of the presence of scale-forming materials. The general classifications of hardness are *temporary* and *permanent.*

Waters containing only bicarbonate compounds of magnesium and calcium are classed as having temporary hardness. The bicarbonate compounds are formed by waters impregnated with carbon dioxide dissolving, as they seep through limestone, either or both magnesium carbonate, $MgCO_3$, or calcium carbonate, $CaCO_3$. The CO_2 in solution has the property of a weak acid and the resulting bicarbonates $MgH_2(CO_3)_2$ and $CaH_2(CO_3)_2$ are produced. Water alone is not a solvent for either $MgCO_3$ or $CaCO_3$.

The application of heat to waters having temporary hardness will cause the CO_2 content of the bicarbonates to be driven off, with the result that the carbonates are precipitated. This process is nearly complete at a temperature of 212 deg. fahr. so that waters with only temporary hardness can be softened by heating. The resulting sludge caused by the precipitation may be blown out of the boiler or removed from the feedwater heater. The degree of softening is affected by the average temperature to which the waters are subjected. Heating to a temperature of 210 to 211 deg. fahr. in an open type of feedwater heater may be sufficient. The scales formed by untreated or insufficiently treated waters of temporary hardness are softer and more porous than those produced by permanently hard waters.

Permanent hardness is a term applied to waters having scale-forming materials not precipitated by heat alone at temperatures less than 300

deg. fahr. Permanently hard waters contain magnesium sulphate, $MgSO_4$, or calcium sulphate, $CaSO_4$, or both.

Boiler feedwaters may also contain chlorides and nitrates of calcium and magnesium. Under heat and pressure the chlorides and nitrates are prone to break up to form corrosive acids. Sodium carbonate, Na_2CO_3, sometimes appears free in raw waters and quite often in waters overtreated with soda ash. Many engineers believe that water treatment by the zeolite process produces free sodium carbonate in some waters. Under heat, sodium carbonate is very likely to be decomposed to a hydroxide or caustic soda, NaOH. Overstressed or strained boiler parts have been embrittled by the action of sodium hydroxide, NaOH.

The relative hardness of waters is expressed in terms of equivalent grains of scale-forming material per U. S. standard gallon. The amount of this material in untreated waters may be from 2 to 140 grains per gallon. Waters having less than 8 grains of scale-forming materials per gallon are classified as very good; those which have 20 or more grains per gallon are poor; while those having 30 or more grains per gallon are bad. The equivalent grains of scale-forming materials per gallon of the water are not a complete indication of the desirability of the water for boiler purposes. Not all scale-forming materials are of equal undesirability. Furthermore, some waters either untreated, partially treated, or overtreated, may carry the ions of certain inoffensive salts which are likely to produce destructive combinations within the boiler.

109. Methods of Water Treatment.—Water treatment should be undertaken only after analyses of the water and recommendations for its treatment have been made by an experienced and competent chemist. There is no single prescribed process of treatment or boiler compound that is suitable for all waters. Each case must be considered individually.

Water treatment processes include removal of debris by stationary or moving screens, filtration, separation of mud and silt in settling basins by subsidence, heating, evaporation or distillation, deaeration, treatment with lime, treatment with soda ash, treatment with both lime and soda ash, lime and barium, coagulants, and zeolite softeners. The processes are not all carried out in the same manner; some of the methods are continuous; others are intermittent. Cold water is treated in some cases, while heat is necessary for best results in many methods. The space available for the treating plant may be a deciding factor in the selection of the treating system. The method chosen depends upon the degree of purification necessary as determined from operating conditions, and the economic balance between the cost of treatment and the ultimate return on the investment necessary.

In any event, for the best results the purification of the water should take place outside of the boiler. The primary objects of the purification are twofold:

(1) The removal of suspended and soluble solids.

(2) The removal of gases.

Where the water is treated with some reagent as it is fed to the boiler or where heating in the boiler itself is required to precipitate the suspended solids, all of the sludge material occasioned by either method is held within the boiler. This method necessitates frequent blowing down of the boiler to remove the precipitated matter and to reduce the concentration of the salts held in solution by the boiler water. Frequent blowing down of the boiler results in a waste of heat carried away by the water and the material removed from the mud drum of the boiler. Devices to allow the continuous blowing down of boilers are available; many of these serve as heaters for the incoming feedwater and thus recover a greater part of the heat carried by the water and sludge thrown away.

110. Filtration.—The removal of suspended matter in water is often accomplished by the use of mechanical filters. The water is usually passed downward through layers of finely divided filtering media such as sand, quartz, or gravel. The use of coagulants introduced into the water as it passes into the filter aids materially in the maintenance of a rapid rate of filtration. The common coagulants used are alum, sulphate of aluminum, or iron sulphate. These chemicals when added to the water react with the alkaline salts produced by bicarbonates or carbonates of lime and magnesia to form a bulky gelatinous precipitate of hydrate of aluminum or iron. This gelatinous hydrate forms a blanket in the filtering material which effectively removes small particles of suspended matter. Automatic coagulant feeding

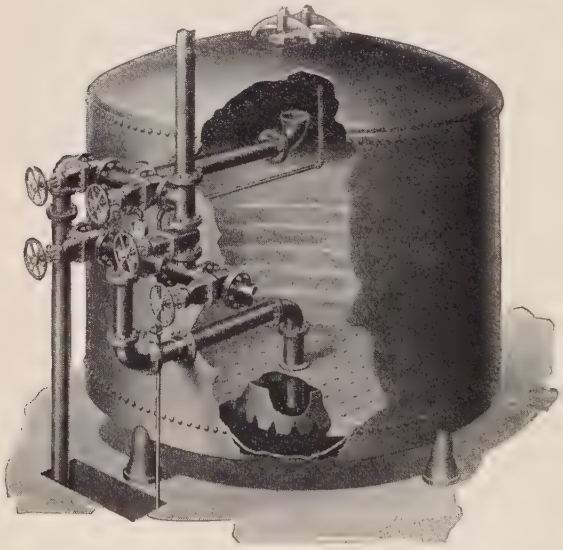


Fig. 82.—Graver vertical pressure filter.

devices are provided for the introduction of the coagulant into the water to be filtered.

The pressure filter is commonly used and may be of either the vertical or horizontal type. The units may be single or in multiple. A Graver vertical pressure filter is shown by Fig. 82. As is common with all pressure filters, this device consists of a closed cylindrical shell containing the filtering material. The water enters the filter body at the top and passes downward through the filtering media and the perforated strainer plate which supports the layers of the filter bed. The filtered water is removed from the bottom of the shell below the strainer plate. Suitable piping connections are provided so that the filter may be backwashed by introducing water under pressure below the strainer plate and forcing it upward through the filter bed to remove entrapped debris.

111. Boiler Compounds.—When composed of the proper ingredients, boiler compounds may prove beneficial. However, the ingredients of the compound must be adapted to the water used as a single formula is not applicable to all waters. Many worthless or even dangerous products are marketed under the guise of boiler compounds. In any event the use of compounds necessitates the removal of the solids from the boiler. Such palliatives do not remove occluded gases from the water.

112. Lime Treatment.—Lime in the form of lime water, $Ca(OH)_2$, is suitable for the treatment of temporary or bicarbonate hardness. The carbonic acid present in the water either in a free state or in bicarbonate combination is taken up and the insoluble precipitate of lime carbonate formed. The process is best carried out in large tanks from which the softened water may be decanted and the sludge removed from the treating plant.

113. Soda Treatment.—The soda process is suitable for waters containing sulphate hardness. Either caustic soda, NaOH, or sodium carbonate, Na_2CO_3, or both may be used. The soda ash, Na_2CO_3, causes the decomposition of the sulphates and the precipitation of insoluble lime carbonates and soluble magnesium carbonates. The addition of lime converts the soluble magnesium carbonate into an insoluble hydroxide. Caustic soda takes up free carbon dioxide and converts magnesium sulphate into insoluble magnesium hydroxide. Tri-sodium phosphate may be used as a reagent for water treatment in place of Na_2CO_3. Tri-sodium phosphate is more expensive than soda ash but the water thus treated has a lower content of caustic material. The latter condition is particularly advantageous in high-pressure boilers.

114. Lime and Soda Process.—This is a common method employed where both bicarbonate and sulphate hardness exist. The lime added

takes up the CO_2 not affected by the soda used in the treatment of the sulphates. Fig. 83 illustrates a Vater hot-process treating plant which is suitable for the use of lime, or soda, or both. The lime and soda reagents are taken from the chemical tanks and the properly proportioned amounts of each are introduced into the raw water as it enters the softener proper. The incoming water mingles with exhaust steam as it enters at the top of the softener and is heated to about 210 deg. fahr. The water slowly passes downward and the precipitated material is deposited at the bottom of the softener shell where it may be removed through the sludge outlet. The treated water is withdrawn above the level at which the sludge collects and is normally passed through a filter before it goes to the boiler feed-pump suction. The use of sodium aluminate facilitates the action of the precipitating reagents and also serves as a coagulant in any lime or soda process of treatment.

115. Hot and Cold Processes.—Treatment with lime or soda as reagents may be accomplished by using either heated or unheated water. In general, heating hastens the process of precipitation and requires smaller tank equipment. Where steam is not available for water heating, the cold process may be employed with larger treating tanks, and at a slower rate.

116. Zeolite Treatment.—Natural or artificial hydrated silicates of aluminum having an alkaline base, have the property of removing calcium or magnesium from water passed through them. These materials have the chemical composition of $2SiO_2 \cdot Al_2O_3 \cdot Na_2O$ and are known as zeolites. Their bases are interchangeable so that the sodium of the zeolite goes into solution as sodium carbonate or sulphate as the calcium and magnesium are absorbed from the water. There are no precipitates or sludges formed in this process of treatment. The zeolite material is regenerated when it becomes inactive by washing it with salt (NaCl) brine. Waters for zeolite treatment must be cool and free from debris, mud, silt, or iron, as these materials clog or coat the particles of zeolite. Where the water is dirty, acid, or where iron exists, the water must have prior treatment and purification. A sectional view of a Graver vertical zeolite softener is illustrated by Fig. 84. The softening equipment consists of two tanks as shown; the large shell with the dished heads is the softener proper; the water to be softened enters at the bottom of the shell and passes upward through the strainer plate. The strainer plate serves to distribute the water, and supports the minerals held within the softener. The layers of materials from the strainer plate upward are: (1) coarse gravel, (2) medium sized gravel, (3) fine gravel, (4) a thicker layer of quartz, (5) the zeolite bed. The flow of water is upward through the filter beds and the zeolite materials. The

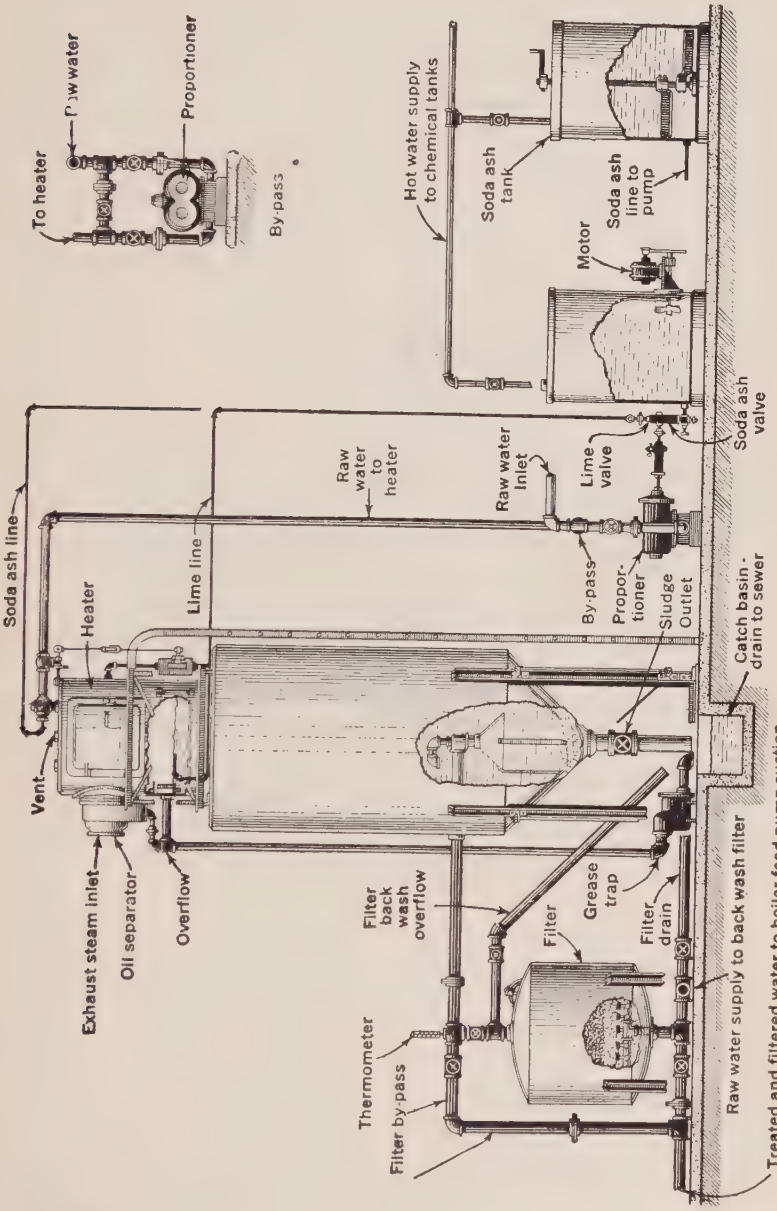


FIG. 83.—Water hot-process water-softening system.

softened water is taken from near the top of the tank at a distance above the zeolite bed to prevent loss of the zeolite materials. The open tank at the right of the figure is the brine tank and is used when the softener is reconditioned. The proper control valves for normal operation and for regenerating conditions are shown on the front of the apparatus.

As the process is one of an exchange, zeolite-treated waters carry a considerable quantity of sodium salts in solution. Waters thus impregnated are likely to cause foaming and priming in the boiler and they may also cause caustic embrittlement of the boiler plates.

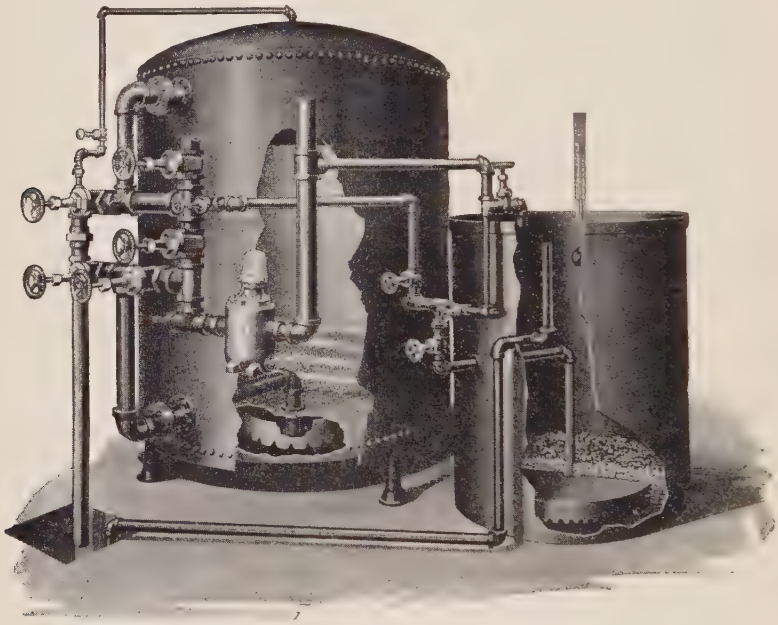


FIG 84.—Graver vertical zeolite water-softener.

Certain waters should be first treated with lime to remove the carbonates and then put through the zeolite material to remove the permanent hardness or sulphate. The lime zeolite-process removes the danger of caustic embrittlement. Zeolite and untreated waters containing free sodium carbonate, Na_2CO_3, may be treated with an acid to reduce their causticity. The water should always be left slightly alkaline, and the treatment must be done under the direction of someone capable of handling such work, otherwise, adding an acid to the feedwater may be very dangerous.

117. Evaporators.—Nearly pure water can be obtained by distillation of the feedwater supply. In plants where surface condensers are

used and their condensate returned to the boilers, the distilled water to be provided is only that necessary to make up the quantities of water disappearing by leakage and unavoidable losses. Distillation of the makeup water supply only is commercially feasible. The use of distilled water almost eliminates the formation of scale and other feedwater difficulties in boiler operation. Even where distilled water is provided small amounts of scale-forming materials may be carried into the boilers by condenser leakages and by solids thrown over in the distilling equipment. The amounts of these solids are usually small. In the treatment of sea water with its large and varied mineral contents, distillation is the only satisfactory process.

The distilled water necessary is obtained by the use of evaporators. Various types of this equipment are built to operate using steam as a heating medium, at high, moderate, low, or less than atmospheric pressures. The vapor produced by the evaporator may be condensed in the feedwater heaters by the incoming feedwater or in separate evaporator condensers using the feedwater to be heated as the cooling water.

Evaporators may accomplish the vaporization by any one of three methods, depending upon their construction. These types and methods are:

(1) *Flash type*, hot water is injected into a chamber under vacuum where vapor is produced.
(2) *Film type*, water in a thin film is passed over steam-filled tubes.
(3) *Submerged type*, steam-filled tubes are submerged in the water to be evaporated.

Evaporators are also further designated as being single or multiple effect units, depending upon the number of evaporating stages through which the water is passed. Thus a single-effect unit is one in which the vaporization is completed in a single coil or stage. A double effect is secured by condensing the vapor from a single-effect unit in the coils of a second evaporator. The second evaporator operates at a lower temperature and from it the vapor passes to the feedwater heater or the evaporator condenser. Multiple-effect evaporators utilize more of the heat of the steam supplied to them but they do not give any greater evaporating capacity per square foot of heating surface than single-effect evaporators produce.

The low-pressure evaporators usually exhaust to open types of feedwater heaters at about atmospheric pressure while the high-pressure types of evaporators exhaust to special evaporator condensers.

A Wheeler film evaporator is illustrated by Fig. 85. The steam pressures maintained in the nests of return bend connected tubes may

be at a pressure greater or less than atmospheric. In either case the condensed steam and the non-condensable gases of the steam are removed from the vent manifold. The water to be evaporated is sprayed continuously over the tubes by the action of a centrifugal recirculating pump forcing the water through nozzles above a perforated distributing tray. The water level in the shell of the evaporator is always held below that of the lower tubes. High rates of heat transmission are obtained with small temperature differentials. The washing action of the water over the tubes tends to prevent the tubes from becoming scale coated.

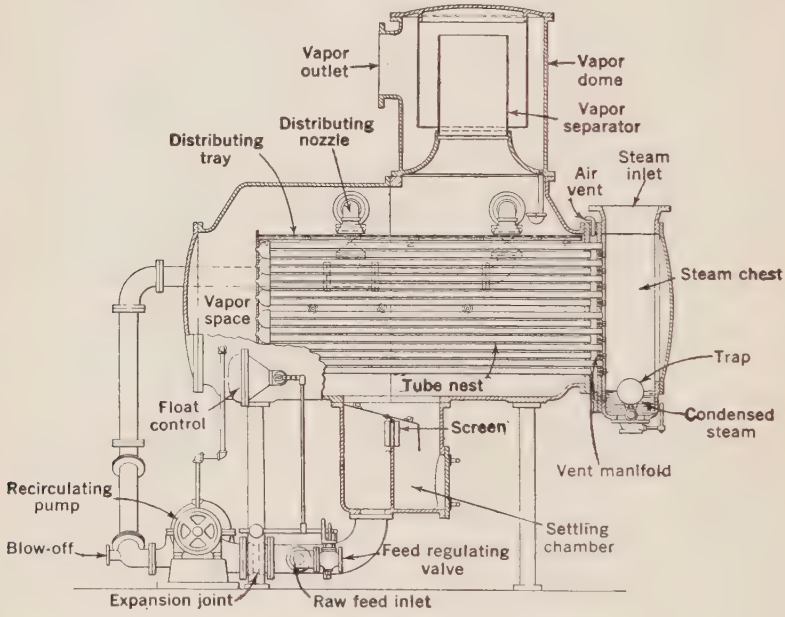


FIG. 85.—Foster Wheeler film-type evaporator.

Scale is also removed from the tubes by reversal of the vapor and water flow, causing unequal expansion rates of the tubes and the scale. As a result, the scale falls from the tubes to the settling chamber. The vapor produced in the evaporator is carried away to a suitable evaporator condenser or heater.

118. Deaerators.—Air, oxygen, carbon dioxide, or other occluded gases carried by water may be active in producing corrosion within boilers, piping, heaters, and economizers. Mechanical devices used to liberate these gases are known as deaerators. In any event the water is boiled within the deaerator and in the process, air, CO_2 and other gases are separated from the vapor as it condenses. Steam is used as the heating

medium in deaerators at pressures ranging from high to those less than atmospheric.

The open type of feedwater heater inherently serves as a deaerator provided that the water is heated to a temperature high enough to drive off the gases held within the water. The escape of the gases takes place through the vent of the heater. Several manufacturers equip either the open or the closed types of feedwater heaters with deaerating attachments, thus making a single unit serve as both a heater and a deaerator.

Deaerators are made in several styles depending upon the nature of the operating conditions of the plant. Fig. 86 shows a diagrammatic

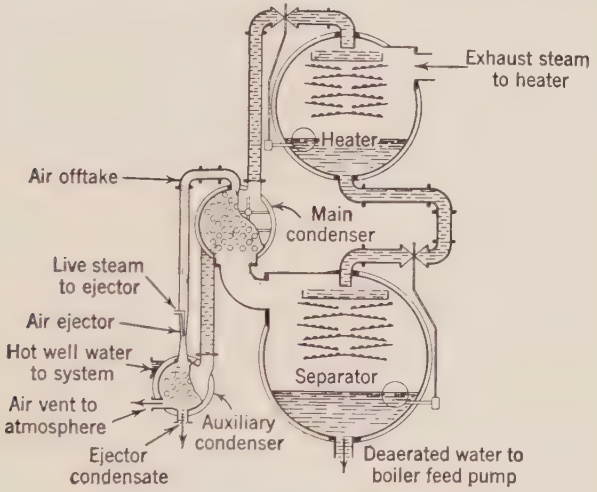


Fig. 86.—Elliott type "E" deaerator.

sketch of the cycle of operation of a Type E Elliott deaerator. Water from the main supply or hot well is passed through the auxiliary and main condensers of the deaerator, as it flows to the heater. Exhaust steam supplied to the heater produces the desired water temperature. From the heater the water is passed to a separator which is under a pressure less than atmospheric. The vacuum is always great enough so that the temperature corresponding to the absolute pressure in the separator is less than that of the water. As a result of the reduced pressure the water in the separator is boiled violently and some vapor, together with the dissolved gases is thrown off. The water cascades over the separator trays and is removed from the bottom of the chamber. The vapor and emitted gases pass to the main condenser where the vapor is reduced to water and the non-condensable gases are removed by a steam-actuated air ejector. The heat of the vapor and the heat of the

air ejector exhaust are recovered by the incoming water as it passes through the main and the auxiliary condensers.

119. Feedwater Heating.—The economical production of power necessitates the utilization of a maximum amount of the heat that may be obtained from a fuel. In steam boiler operation the flue gases always leave the last pass of the boiler setting at a temperature well above that of the saturated steam and the water within the boiler. In many but not all plants a part of the heat of the escaping flue gases may be profitably employed to heat the boiler feedwater or to preheat the combustion air.

A considerable quantity of exhaust steam, with a large heat content, is available in plants having the prime movers operating non-condensing. Steam-engine- or turbine-driven auxiliaries such as draft fans, boiler feed pumps, electrical exciter units, air compressors, etc., provide additional exhaust steam for heating purposes. A part or all of such exhaust steam may be profitably used for water heating.

Some large central stations operate with all of the auxiliaries electrically driven by a house turbo-generator exhausting into a condenser using feedwater for cooling water. The exhaust supply may be inadequate so that high-pressure steam or steam bled from the various turbine stages is used with economical results for feedwater heating.

The distinct advantages that may accrue from feedwater heating are enumerated by the following items:

(1) Reduction of boiler strains.
(2) Utilization of heat otherwise likely to be lost.
(3) Partial purification of untreated feedwater.
(4) Closer approximation of ideal thermal efficiencies for the plant by bleeding steam from turbine stages.
(5) Relief of the last stages of steam turbines from great volumes of low-pressure steam by stage bleeding to feedwater heaters.
(6) Increased boiler capacity.

120. Economy of Feedwater Heating.—Many boilers do not show an increase of efficiency when supplied with heated water; others show an efficiency increase or decrease due to the change in the rate of heat transfer. In any event the saving due to feedwater heating appears in the reduced fuel consumption and the overall efficiency increase for the plant. The amount of the fuel saving is dependent upon the boiler efficiencies, the initial and final temperatures of the feedwater, and the total heat content of the steam produced, where waste heat is used for water heating. The saving effected is a variable dependent upon the operating conditions of the plant.

The fuel saving due to feedwater heating may be expressed by:

$$S = \frac{100\left(\frac{t_2 - t_1}{e_b}\right)}{\frac{H - (t_1 - 32)}{e_b}} = \frac{(t_2 - t_1)100}{(H - t_1 + 32)} \quad . \quad . \quad . \quad . \quad (95)$$

where S = fuel saving, per cent.
H = total heat of the boiler steam, B.t.u. per lb.
t_1 = initial temperature of the feedwater, deg. fahr.
t_2 = final temperature of the feedwater, deg. fahr.
e_b = overall efficiency of the boiler unit.

Example.—A small plant operates with 60 per cent boiler efficiency, a steam pressure of 175 lb. per sq. in. absolute, and 50 degrees of superheat. If the feedwater is heated from 60 to 205 deg. fahr., what fuel saving will be effected?

Solution:

$$S = 100\frac{(205 - 60)}{(1226.7 - 60 + 32)} = 12 \text{ per cent.}$$

121. Feedwater Heater Classifications.—Divisional groupings of feedwater heaters may be made with reference to:

(1) Heat supply.
(2) Method of heat utilization.
(3) Steam pressure.
(4) Steam-flow arrangements.

When the heat supply is from hot flue or waste gases, a water heater is designated as an *economizer*. When the heat utilized is derived from exhaust, bleeder, gland, jet, or live steam sources the term *feedwater heater* is applied to the equipment. Feedwater heaters of the *open type* utilize the heat of the steam by the direct contact of the steam and water as the mixture is made. Open heaters operate at pressures ranging from atmospheric to 15 to 20 lb. per sq. in. gage. The steam and the water are at the same pressure.

In the *closed type* of heater the heat of the steam is transmitted through metal walls. The steam and water are not in direct contact. Closed heaters may operate at any steam pressure ranging from less than atmospheric to high. In closed heaters the water and steam pressures are not usually the same.

Heaters operating at pressures less than atmospheric are termed *vacuum* or *primary*. When the steam pressure is atmospheric or only slightly above atmospheric, heaters are classed as *atmospheric or secondary*. The *pressure* type of heater as the name implies may carry any reasonable pressure above that of the atmosphere.

When the steam is drawn into the heater by the partial vacuum produced by steam condensation, the heater is of the *induced type*. Steam is taken into the heater only as it is required. In the *through* type of heater the steam is forced through the heater irrespective of the amount of steam condensed or the requirements for steam.

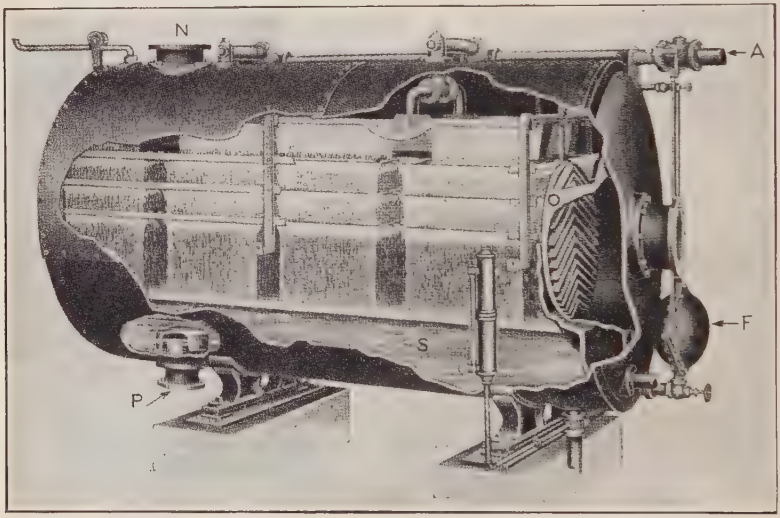


FIG. 87.—Hoppes horizontal open-type feedwater heater.

122. Open Feedwater Heaters.—These heaters may be built as either horizontal or vertical units. Either type requires the following principal parts, an enclosing cast-iron or steel shell, a steam and oil separator at the steam inlet, water trays or pans over which the feedwater overflows or is sprayed, steam and water controls, oil trap, excess steam outlet, boiler feed pump connection, blow-off connections, and air and steam vent. A small amount of water storage space is always provided at the bottom of the heater and in some cases a water filter is located within the heater shell.

A Hoppes horizontal open type of heater is illustrated by Fig. 87. The water enters the heater from the line *A* and the amount of water entering the heater is controlled by a float located in the box *F*. The water is supplied to the uppermost cast-iron pan from which it over-

flows, falling successively into the pans beneath and finally to the storage space *S*. Steam which has passed over the oil separator *O* at the inlet fills the heater and surrounds the trays. The heat of the steam is given up to the flowing water. The trays can be taken out for the removal of precipitated scale deposits. Any excess steam supplied and gases liberated escape at the outlet *N*. The outlet *P* is placed in the bottom of the steel shell. This heater serves to deaerate the water, however, the effectiveness of the degasification process is dependent upon the water temperatures attained.

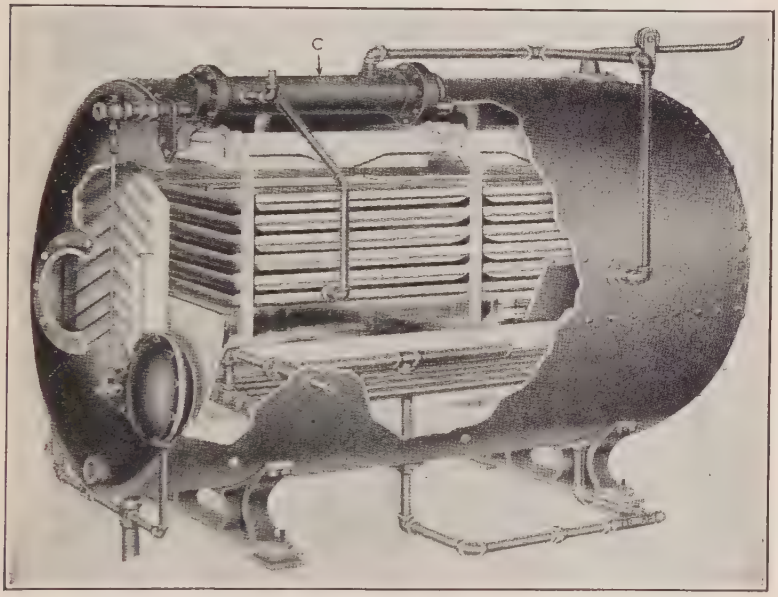


FIG. 88.—Hoppes horizontal deaerating feedwater heater.

The Hoppes open heater shown by Fig. 88 is similar to that illustrated by Fig. 87. This heater includes more positive deaerating equipment. A steam coil is located in the water storage space. Steam, at 5 lb. per sq. in. gage or more pressure, is supplied to the coil and the water within the heater is made to boil. The vapor produced together with the gases liberated is carried to a condenser *C* cooled by the incoming water. The non-condensable gases escape through a vent from the condenser and the condensed vapor is returned to the heater.

A Cochrane vertical open-type heater with cast-iron shell is shown by Fig. 89. The incoming steam passes over a fluted oil separator and fills the heater body. The water to be heated enters at the top of the shell and falls as a spray into the steam which fills the heater. As the water

passes downward it is directed over inclined and staggered cast-iron trays. The trays are both serrated and perforated, thus providing surface for breaking up and heating the water as well as for the liberation of gases. The trays are removable so that scale deposits may be taken out. The bottom of the heater serves as a storage tank and the amount of water admitted to the shell is under the control of a float-actuated valve in the supply line. The heater shown has piping connections for condensation

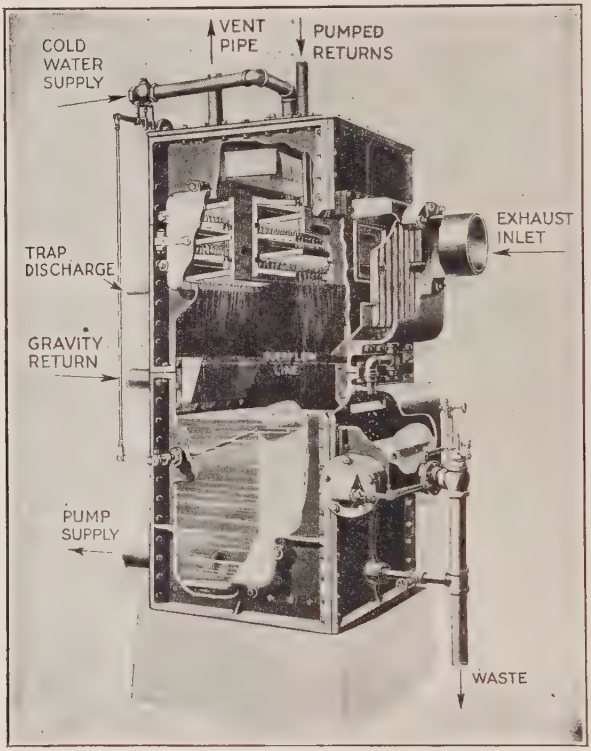


Fig. 89.—Cochrane vertical open-type feedwater heater.

returns either by pump or gravity. Filtering material is provided in the water storage space. When the heater is operated under steam pressures greater than atmospheric the vent pipe should be connected to a condenser. The use of the condenser will permit the escape of liberated gases and allow the return of water which otherwise would escape as steam. Floating debris is discharged through a trap which also eliminates the oil removed by the oil separator. Cochrane heaters are also built with special deaerating attachments.

All heaters of the open type must be located well above the pump suction so that water from them will flow freely to the pump by gravity, as pumps will not lift hot water by suction. Open-type heaters may be operated with considerable steam pressure. In any event either the heater or the steam supply line should be provided with relief connections so that excessive and dangerous steam pressures will not be applied to the heater shell.

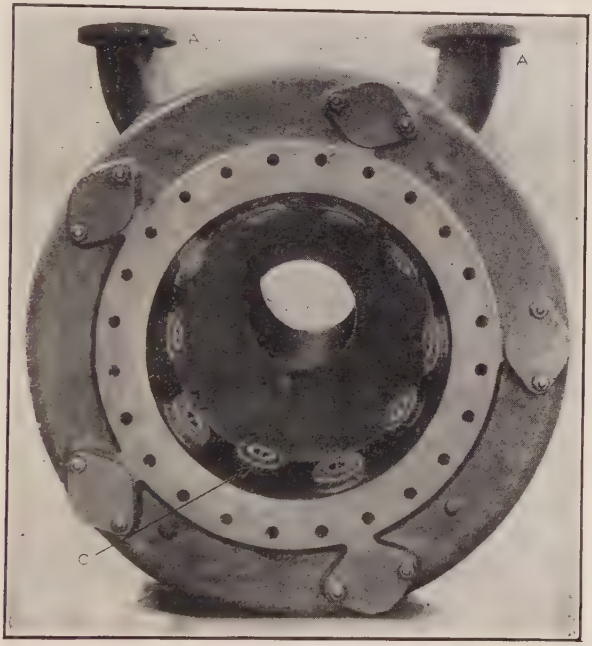


Fig. 90.—Cochrane jet heater, interior view.

123. Jet Heaters.—These heaters employ more or less the principle of the jet condenser. As the steam and the water to be heated are mixed together, they are considered under the scope of open heaters.

An interior view of a Cochrane jet heater with two sets of spray nozzles *C* is shown by Fig. 90. The water inlets of this heater are located at the points marked *A*. Fig. 91 illustrates the exterior view of a Cochrane jet heater with duplex water belts. The water inlets are designated by *A* and the steam inlet by *B*.

The water to be heated is sprayed through the nozzles into a chamber filled with steam. The steam may be under pressure or a vacuum. The pressure of the steam may be rapidly changed from one condition to the other with variations in the flow of the feedwater or the supply of the exhaust or bleeder steam.

Thorough heating of the water is accomplished as the water in small drops falls through the steam and passes to the hot-water storage reservoir or hot well. These heaters have no particular advantages over the ordinary open types of heaters in sizes of less than 5000-hp. capacity. Adjustments of the water-spray nozzles must be made as the demand for water increases or decreases.

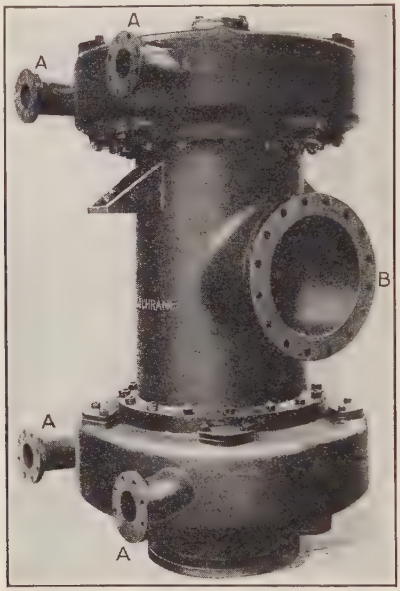


Fig. 91.—Cochrane jet heater, exterior view.

124. Water Temperatures of Open Heaters.—Theoretically the heat given up by the steam should equal the heat absorbed by the water.

$$W_s(H - h_2) = W_w(h_2 - h_1) \quad . \quad . \quad . \quad . \quad . \quad (96)$$

where W_s = weight of steam condensed, lb.
H = total heat of the steam condensed, B.t.u. per lb.
h_1 = heat of liquid of inlet water, B.t.u. per lb.
h_2 = heat of the liquid of outlet water, B.t.u. per lb.
W_w = weight of water heated, lb.

Expressed in terms of temperatures, assuming that the specific heat of the water is 1, equation 96 becomes

$$W_s(H - t_2 + 32) = W_w(t_2 - t_1) \quad . \quad . \quad . \quad . \quad . \quad (97)$$

but t_2 can never exceed the steam temperature. t_1 and t_2 equal the initial and final water temperatures, deg. fahr. In actual operation there are certain inevitable heater losses. Consequently the value of W_s will always be greater than the amount calculated from the equation for a given quantity of water raised from a temperature of t_1 to t_2, by steam of a given pressure and quality. The heater losses may range from 5 to 10 per cent of the heat content of the steam actually used in the heater.

The weight of water leaving the heater is the sum of the weight of water supplied and the weight of steam condensed in the water.

125. Closed Feedwater Heaters.—The steam and water are never in contact in the closed type of heater. Consequently they may be operated at pressures ranging from high to very low. The essential parts of a closed heater are a shell and the enclosed tubes or coils through which either steam or water is circulated. Heaters having the water inside the tubes or coils are called water-tube heaters; if the steam is confined within the tubes the designation of steam tube heater applies. The water-tube heaters are those most commonly used: the steam and water flow may be in the same or in opposite directions in either heater giving either parallel or counterflow conditions. In single-flow heaters the water flows in one direction only; multiflow heaters have the water flowing back and forth in various nests of the tubes. Closed heaters are ordinarily placed in the line between the boiler feed pump and the boilers.

Fig. 92 illustrates a Schutte and Koerting closed heater that may be installed in either a vertical or horizontal position. This heater is of the multiflow water tube type. The straight water tubes are packed into tube sheets at the top and bottom of the enclosing steel shell. The lower tube sheet is not attached to the shell so that provision for expansion and contraction of the tubes is made.

Water enters the top header which is divided into compartments. The feedwater flows into one compartment and passes downward through one-fourth of the tubes to a compartment of the bottom header fastened to the lower tube sheet. The water makes two passes downward and two upward through the four-tube nests before leaving the heater. Heating steam enters the top of the shell and surrounds the tube nests. The condensation is removed through a drain connection at the bottom of the shell. The assembled tube nests may be removed for inspection and cleaning without disconnecting the steam piping.

A Patterson-Berryman bent-tube type of closed feedwater heater is shown by Fig. 93. This heater consists of a cast-iron or steel shell containing a nest of inverted U-shaped circular copper tubes. The bent-tube construction takes care of strains produced by expansion and

contraction of the small water passages. The heating surface is divided into from two to seven rows of tubes, depending upon the capacity of the heater. The water in the tubes passes through the surrounding steam as many times as there are rows of tubes. The small smooth copper tubes permit a rapid transfer of the heat of the steam to the water. The lower part of the heater serves as a settling chamber, from which precipitated impurities may be removed.

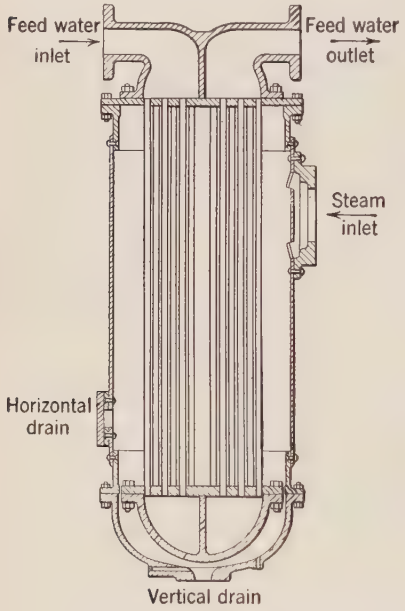


Fig. 92.—Schutte and Koerting closed-type feedwater heater.

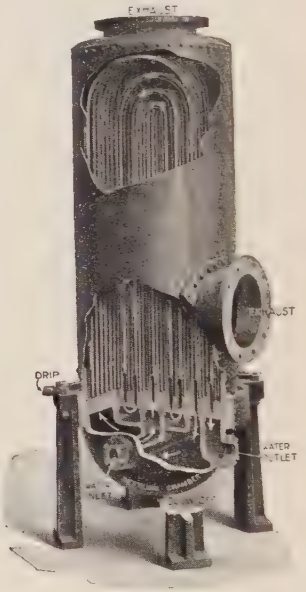


Fig. 93.—Patterson-Berryman feedwater heater.

126. Heat Transfer in Closed Heaters.—The principles of heat transmission in closed feedwater heaters and surface condensers are essentially alike. If liberal factors are allowed in the design of closed heaters the laws of each may be taken as the same.

The amount of heat given up in the heater per square foot of heating surface is dependent upon the velocity of the water through the heater and the mean temperature difference between the steam and the water. An increase in either or both of these items will increase the amount of heat taken up by the water. Sufficient time must also be allowed in order that the temperature rise may be a maximum under the conditions of operation. In general, all other conditions being the same, a heater having a large number of passages of small cross-sectional area

will transmit more heat than one having a few passages of large cross-sectional area.

The amount of heating surface necessary to heat a given quantity of water through a desired temperature rise is expressed as

$$A = \frac{W_w c_{pw}}{K} \log_e \left(\frac{t_s - t_1}{t_s - t_2} \right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (98)$$

where A = total heating surface, sq. ft.
c_{pw} = specific heat of the fluid heated.
K = coefficient of heat transmission, B.t.u. per sq. ft. of surface per hour, per deg. fahr.
t_1 = initial temperature of the feedwater, deg. fahr.
t_2 = final temperature of the feedwater, deg. fahr.
t_s = temperature of the steam supplied, deg. fahr.
W_w = weight of water heated per hr., lb.

Equation 98 may also be written

$$A = \frac{W_w c_{pw}}{K} 2.302 \log \left(\frac{t_s - t_1}{t_s - t_2} \right) \quad . \quad . \quad . \quad . \quad . \quad (99)$$

The values of K may range from 120 to 1200 B.t.u. per sq. ft. per hour per one deg. fahr. temperature difference. Average values of K depending upon the heater type, the steam and water temperature differences, and the velocity of the water flow, range from 200 to 600 B.t.u.

127. Relative Merits of Open and Closed Heaters.—Professor G. F. Gebhardt in "Steam Power Plant Engineering," has summarized the following comparisons:

OPEN HEATER	CLOSED HEATER
Efficiency	
With sufficient exhaust steam for heating, the feedwater may reach the steam temperature. Scale and oil do not affect the heat transmission.	The maximum water temperature will always be 2 or more degrees less than the steam temperature. Scale and oil deposits reduce the amount of heat transmitted.
Pressures	
Usually not much above atmospheric.	Water pressure slightly greater than the boiler pressure when placed on the discharge side of the pump.

OPEN HEATER	CLOSED HEATER
Safety	
Failure of the back pressure valve to operate may result in a heater explosion.	Safe at almost any pressure likely to be encountered.
Purification	
Oil must be removed from the exhaust steam supplied, otherwise the feedwater supply will be contaminated.	Oil and the feedwater are not in contact. Scale is difficult to remove.
Scale and other impurities precipitated readily removed.	Occluded gases not removed unless vented to a region of lower pressure.
Liberated gases removed if the heater is properly vented.	
Location	
Above the pump suction on the suction side always.	Anywhere on the discharge side of the pump.
Pumps	
Two required with the supply under suction.	One cold water pump is required.
Adaptability	
Gravity flow return of condensate.	Vacuum and primary heaters. Stage bleeding.

128. Economizers.—Water-filled pipes, placed between the last pass of a boiler setting and the chimney, are used to preheat boiler feedwater by absorption of heat from the flue gases. The assembly of water-filled pipes is known as a *fuel economizer*. The use of water heaters of this type is justified only in plants large enough to show a profitable saving after the interest, depreciation, and maintenance charges for the economizer are paid. Economizers increase the resistance offered to the flow of flue gases and also reduce the flue gas temperatures. Consequently, the natural draft provided by a chimney is often inadequate and induced draft must be maintained. The additional costs incurred through the use of induced draft must also be given attention, when the possible savings resulting from the use of economizers are to be considered.

Two classes of economizers are in use, those having cast-iron tubes and those having steel tubes. Cast-iron tube economizers may be used with boilers operating at 300 lb. per sq. in. pressure or less. The use of higher steam pressures necessitates the use of steel tubes. In the earlier days of the economizer, cast iron was used exclusively as the best non-corrosive tube material. Oxygen carried by the water passing through steel tubes induces rapid corrosion of the steel. Also the sulphur dioxide, SO_2, of flue gases is changed to H_2SO_4, a corrosive acid, when the flue gas temperatures are reduced so that their steam content is condensed to water. The acid formed induces rapid corrosion on the exterior of steel or wrought-iron tubes. Modern steel economizer tubes may be protected against corrosion within and without by protective coatings, or externally by cast-iron coverings, also by deaeration of the water as a relief from internal corrosion.

129. Cast-iron Economizers.—A Sturtevant cast-iron tube economizer is illustrated by Fig. 94. Plan and elevation views of a layout using this type of economizer are shown by Fig. 95.

Vertical cast-iron tubes are joined into headers or manifolds at the tops and bottoms. Such an assembly forms a section which may be had in various heights and widths. The sections when joined together by connections at both the top and the bottom may be erected to form an economizer of any feasible length. The tube arrangement in the interconnected sections may be staggered as shown by Fig. 94, or the tubes may be directly in front of each other.

The water to be heated is introduced into the wall box or lower section connection at the flue-gas exit end of the economizer. The water is distributed to the various sections and rises through the tubes to the top of the economizer. The water from the various sections is carried through interconnecting parts at the top of the economizer to the flue-gas inlet end, where it is delivered to the boiler supply line.

Mechanically operated scrapers are moved up and down on the tubes to remove soot accumulations. The motion of the scrapers is automatically reversed when the scrapers reach the tops and the bottoms of the tubes. The tube sections are enclosed by metal or brick sides so that the flue gases are forced to pass through the openings between the tubes and also between the tubes and the enclosing side walls.

Provision is always made for bypassing the flue gases beneath or around the economizer sections when the economizer is out of service. Access to the interior of the tubes is secured by removing cover plates from the upper tube headers and handhole cover plates from the various interconnecting parts.

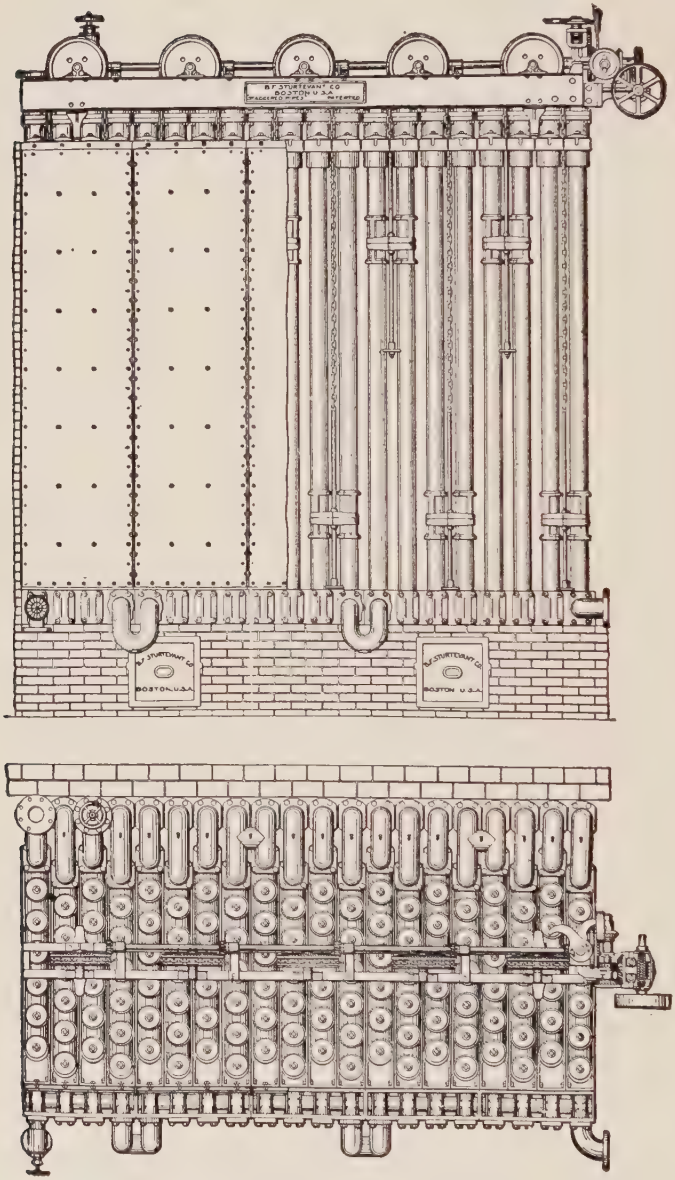


Fig. 94.—Sturtevant cast-iron tube economizer.

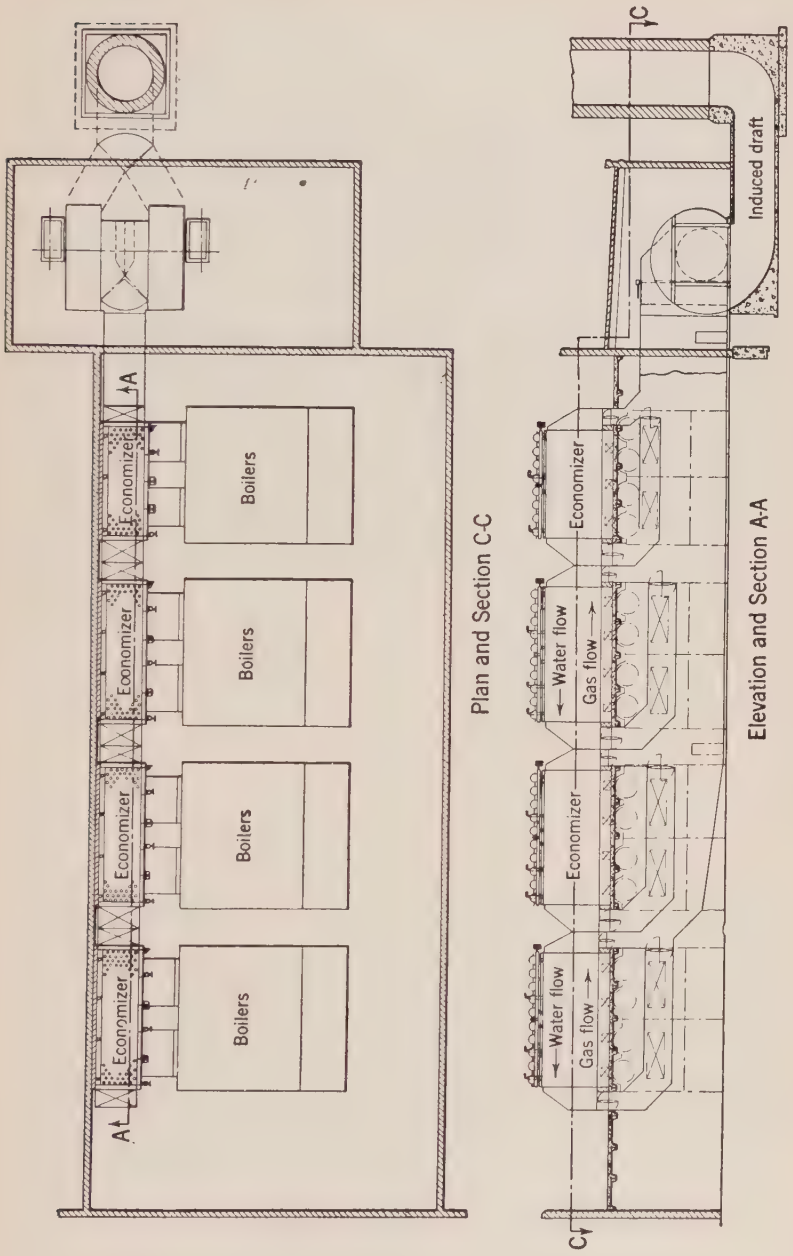


FIG. 95.—Economizer layout.

130. Steel Economizers.—A Sturtevant steel economizer is illustrated by Fig. 96. Steel tubes of about $2\frac{1}{2}$-in. outside diameter are

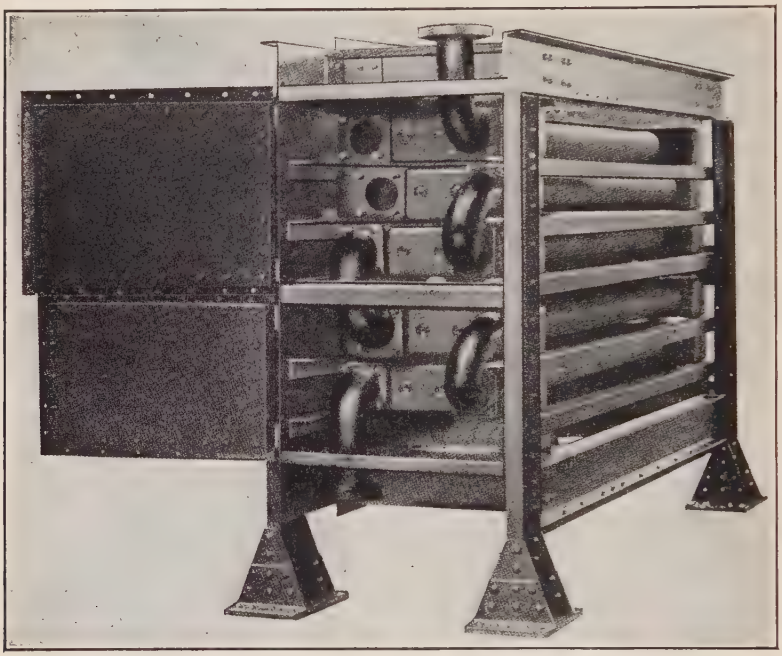


FIG. 96.—Sturtevant steel economizer.

expanded into cast steel headers to form the economizer sections, Fig. 97. The economizer is built up of the assembled sections which are con-

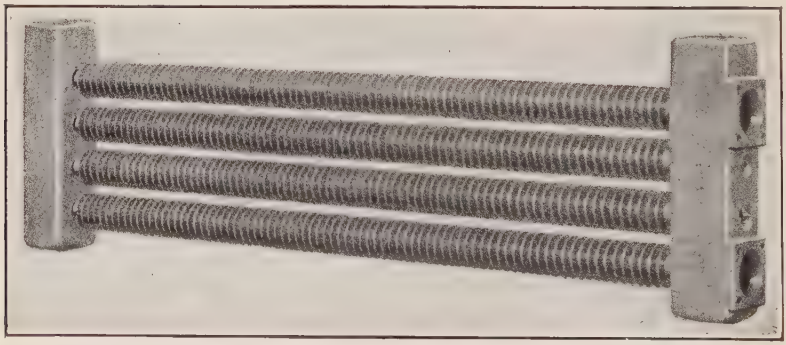


FIG. 97.—Sturtevant steel-tube economizer section.

nected together by U-bends. Each tube carries a number of thin steel disk rings which are uniformly spaced upon the tube and which have

been forced in place under great pressure. The disks make good contact with the tube walls and provide additional surface for the absorption of heat. The entire inner and outer surfaces of the assembled sections are lead coated to prevent corrosion.

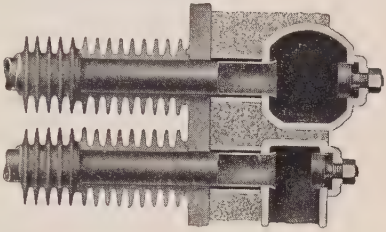


FIG. 98.—Foster steel-tube economizer construction.

The economizer is located within a suitable housing between the last pass of the boiler setting and the chimney or stack. Ash and soot deposits upon the external surfaces may be removed by the use of a steam lance or by the use of permanently installed soot blowers.

The Foster steel tube economizer employs gilled cast-iron rings, Fig. 98, as a protection against corrosive flue gases. The rings have extended surfaces which provide more heat-absorbing surface. The rings are machined to fit together so that all parts of the steel tubes are shielded from the flue gases. Steel is used for all pressure parts. A typical method of installing an economizer of this type is shown by Fig. 99. The water flow through the economizer is opposite to that of the flue gases. Soot blowers are used to remove ash and soot deposits from the heating surfaces of the economizer.

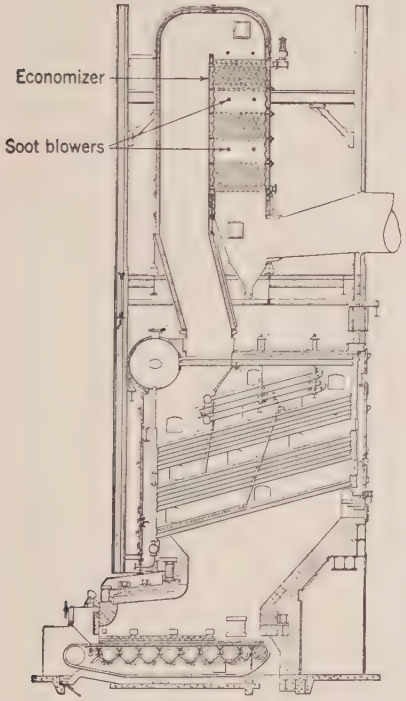


FIG. 99.—Installation of a Foster economizer.

131. Regenerative Feedwater Heating.—Modern power stations operate with their auxiliaries of stoker, pumping, draft, exciter, and lighting equipment driven by

(1) Steam units.
(2) Steam and electrical motors.
(3) Electrical motors.

When all auxiliaries are steam driven a considerable amount of exhaust steam is available in addition to the heat of the flue gases for water heating.

Power for electrical drives may be provided by

(1) A house turbine generator unit.
(2) The bus bars of the main generating units.
(3) A house generator coupled to the shaft of the main unit.

In the latter two cases an independent source of power must be provided for starting the plant. The house turbine generator usually exhausts to a condenser which serves as an open heater. Plants equipped to draw auxiliary power from the main units were so designed with the idea of improving the plant heat balance by regenerative feedwater heating. In any event, by any system of auxiliary drive there may be insufficient exhaust steam for water heating so that steam extraction from the various stages of the turbines may be necessary. It is feasible also to extract steam from the receivers located between the cylinders of compound steam engines.

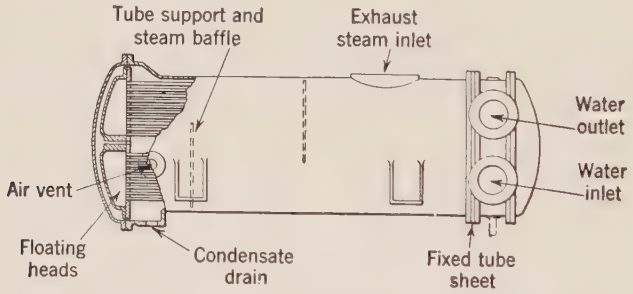


Fig. 100.—Cochrane extraction feedwater heater.

The Rankine cycle, usually taken as the ideal for the production of steam power, is irreversible, and is not the most efficient way of turning heat energy into mechanical energy. The cycle may be made more nearly reversible and the efficiency improved several per cent by heating feedwater with partially expanded steam extracted from a turbine or an engine. Extracted steam has performed a part of the work which it is capable of doing and when used to heat feedwater displaces an equivalent amount of heat in the boiler. Extraction of steam from various points also relieves the low pressure stages or cylinders of a considerable volume of expanded steam.

Extraction from more than one stage is advantageous as it permits a

part of the extracted steam to do more work in the prime mover than if the steam were all taken out at one place.

Furthermore, extraction at several places provides steam at various pressures and temperatures for the progressive heating of the boiler feedwater. Usually not more than three or four stages of heaters are profitable.

At pressures near atmospheric open heaters may be used. Most extraction heaters are of the closed type and operate under steam pressures ranging from high to less than atmospheric. Closed heaters of this type must be able to withstand water pressures considerably greater than the steam pressures carried within the boilers. Ample provision must be made to prevent strains in the heaters by expansion and contraction of the tubes. A Cochrane high-pressure extraction heater with a divided floating tube sheet is illustrated by Fig. 100.

CHAPTER X

DRAFT APPARATUS

CHIMNEYS AND FANS

132. Draft Measurement.—Draft is the difference in pressure between that of the air outside and that of the gases within a furnace, a boiler setting, or a chimney. Such pressure differences are variable in amount and are ordinarily of comparatively small magnitude. The measurements are usually made in terms of the height in inches of a water column exerting an equivalent pressure.

FIG. 101.—Ellison vertical draft gage.

Gages suitable for draft measurements are shown by Figs. 101 and 102. A light oil is used as the gage fluid and the scale divisions are so placed that the readings obtained are in inches of water equivalent. The use of a gage fluid lighter than water permits the graduation marks to be spaced farther apart so that the readings may be made with a greater degree of accuracy. Inclination of the gage scale further increases the distance between graduation marks and the ease with which the indications may be read

An Ellison multiple draft gage of the pointer type is illustrated by Fig. 103. The pressures or the pressure differences to be measured are transmitted to the respective cylindrical bell chambers, which are sealed against air or gas leakage by kerosene gage fluid held in the pan at the bottom of the bells. The hollow bell cylinders move up or down respectively for pressures greater or less than atmospheric. The movements thus produced are transmitted by a straight line motion to the pointer moving over the gage scale. A weight is provided in the linkage mechanism to aid in counterbalancing the gage hand and bell.

FIG. 102.—Ellison inclined draft gage.

133. Draft Requirements.—Draft is required to produce the flow of the air necessary for combustion through the fuel bed and to promote

the flow of gases from the furnace through the boiler setting and smoke connections into and out of the chimney or stack. The resistances offered to the flow of flue gases vary with (1) the type of boiler, (2) the type of the boiler setting, (3) the arrangement of the baffles within the setting, (4) the length and design of the breeching and smoke connections to the chimney, (5) the accumulations of soot and ashes in the setting and the smoke connections, and (6) the rate at which the boiler is being driven. Information relative to the draft losses through various boilers under operating conditions should be obtained from the builders. The resistances offered to the flow of air supplied to the boiler furnace include air-duct friction and the opposition to air flow produced by the fuel bed. The size of the fuel used, the depth of the fuel bed, and the rate of fuel consumption affect greatly the pressure difference necessary to promote the flow of air into a furnace. Draft requirements[1] for two distinct operating conditions are given by Table IX and Fig. 104.

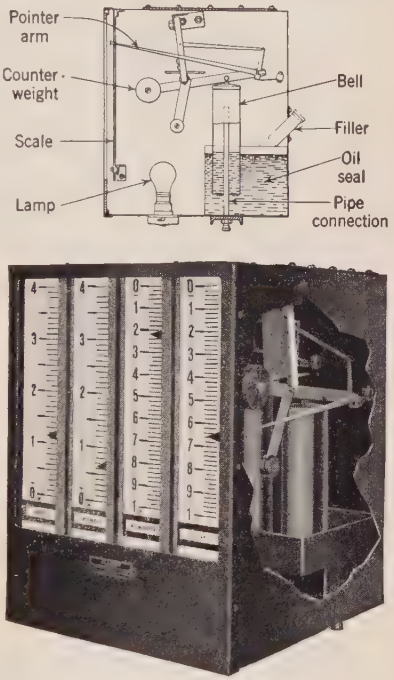


FIG. 103.—Ellison pointer draft gage.

Anything which offers resistance to the flow of air into and through the fuel bed, and the flow of flue gas from the furnace to the chimney increases the amount of draft that must be available. No fixed amount can be applied to all installations, as each boiler plant must be considered individually.

134. Draft Production.—Draft is produced by either of two distinct methods, and is classified as *natural* or *mechanical*.

135. Natural Draft.—The pressure difference thus secured is produced solely by the use of a stack or chimney. The magnitude of the force of draft is dependent upon the height of the chimney top above the level of the furnace grates, and the average temperature difference between the flue gases within the chimney and the outside air. Weather

[1] "Furnace Operation Shown by Draft Measurement." *Power Plant Engineering*, January 1, 1928, p. 15.

variations, boiler-operating conditions, and the chimney height have a marked effect upon the amount of draft that may be secured by the use of a particular chimney.

TABLE IX

Typical Conditions in Natural Draft Furnace

Draft Required in Furnace for Various Rates of Combustion		
Combustion Rate per Square Foot of Grate in Pounds per Hour	Draft in Inches of Water	
	For Bituminous	For Anthracite
5	0.02 to 0.04	0.06 to 0.16
10	0.05 to 0.10	0.16 to 0.40
15	0.07 to 0.15	0.30 to 0.75
20	0.10 to 0.25	0.45 to 1.25
25	0.12 to 0.32	0.65
30	0.17 to 0.45	0.90
35	0.21 to 0.58	
40	0.27 to 0.75	
45	0.32 to 0.95	

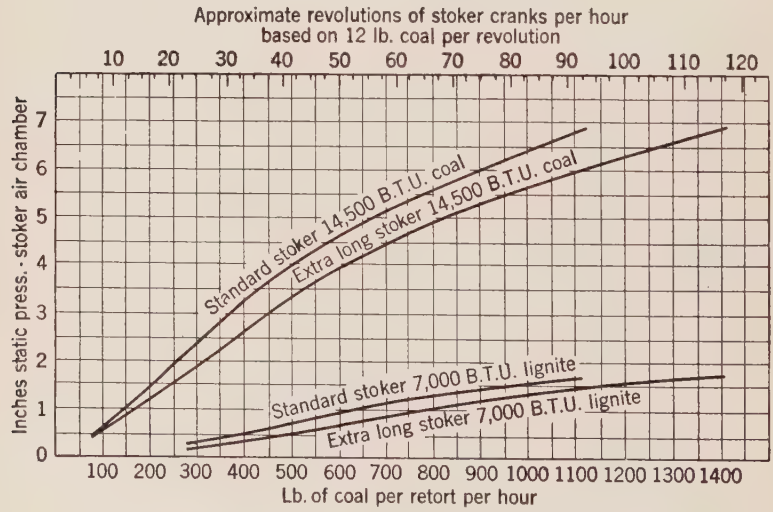

Approximate revolutions of stoker cranks per hour
based on 12 lb. coal per revolution
10 20 30 40 50 60 70 80 90 100 110 120
Inches static press. - stoker air chamber
0 1 2 3 4 5 6 7
Standard stoker 14,500 B.T.U. coal
Extra long stoker 14,500 B.T.U. coal
Standard stoker 7,000 B.T.U. lignite
Extra long stoker 7,000 B.T.U. lignite
100 200 300 400 500 600 700 800 900 1000 1100 1200 1300 1400
Lb. of coal per retort per hour

Fig. 104.—Approximate forced-draft pressures for underfeed stokers.

Natural draft does not require the expenditure of mechanical power in its production and is suitable for small plants when the load conditions are not extremely variable, or where the boilers are not required to develop great capacities quickly.

136. Chimneys and Stacks.—Either of these structures is used with all draft systems for conveying the smoke or flue gases to an elevation above that of surrounding objects and buildings. Structures for the disposal of flue gases or the production of natural draft, when constructed of brick, concrete, or radial brick, are known as chimneys; while those built of sheet steel or steel plates are designated as stacks. Stacks built of the lighter sheet metal require the use of supporting guys, while those built of heavier steel plates may be made self-supporting. An illustration of a typical radial-brick chimney is shown by Fig. 105.

Brick and tile chimneys are likely to have leakages due to poor bonding of the materials or cracks due to settlement of the structure. Concrete chimneys are also subject to cracking. Air leakages into a chimney producing natural draft reduce the amount of the draft available at the smoke connection. Steel stacks, unless kept carefully painted, are subject to the corrosive action of the air and weather. Unlined steel stacks are also corroded by the constituents of the flue gases, so that the life of an unlined stack is much shorter than that of a chimney built of bricks, radial bricks, or concrete.

137. Chimney Height and Area.—Numerous empirical rules have been developed for the calculation of the internal dimensions of chimneys and stacks used for the production of natural draft. The empirical sizes thus obtained may prove adequate in some cases and inadequate in others. A theoretical method of chimney design, in the case of natural draft, is based on an estimation of the probable total-draft requirements, the probable temperatures within and without the chimney, and the volume of the flue gases to be handled per hour.

Natural draft is produced by the difference in the weights of a column of hot gas within the chimney and that of a cold air column of equal height and cross-sectional area without the chimney. Therefore, the higher the chimney, or the greater the temperature difference, or both, the greater is the force produced.

The height of a chimney necessary to produce a given theoretical draft at its base may be calculated in the following manner. A chimney AB, Fig. 106, has a height of h feet and is filled with hot flue gases whose average temperature is t_c. Outside the chimney is a column of cool air whose height and cross-sectional area are equal to those of the chimney and whose average temperature is t_a deg. fahr. The outside air column

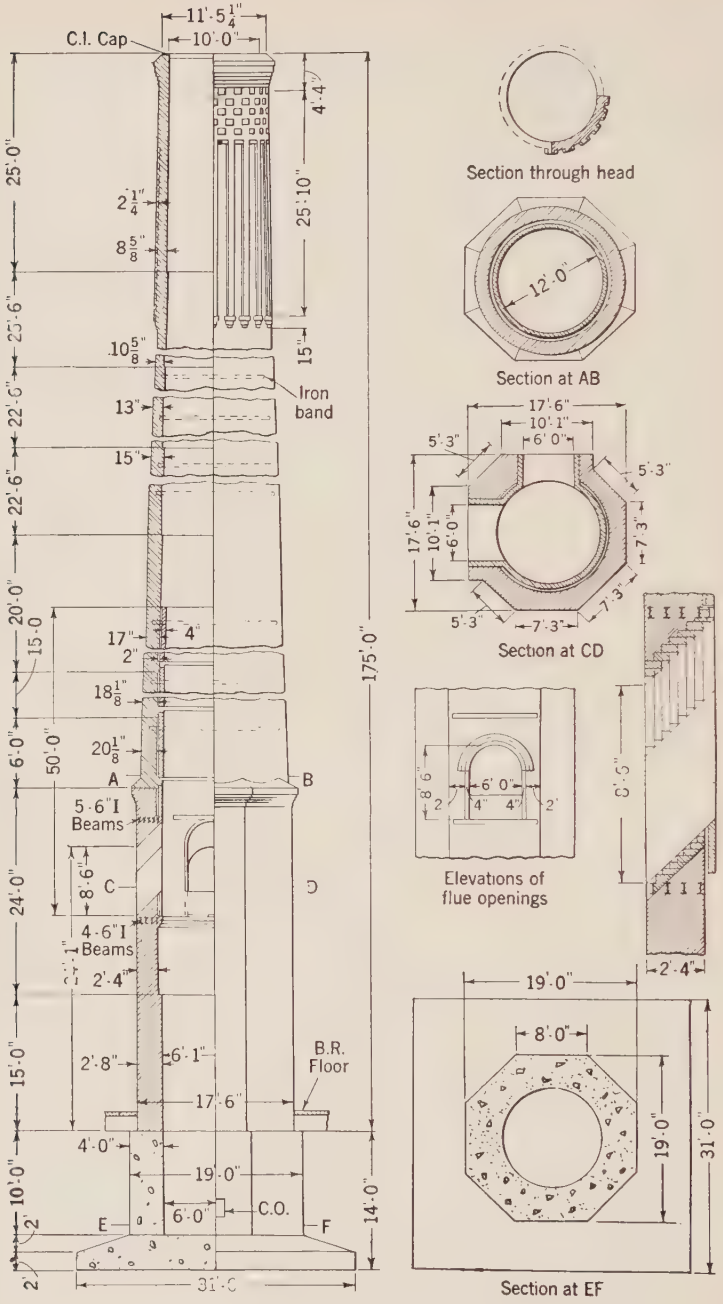


FIG. 105.—Radial-brick chimney, University of Illinois power plant.

is heavier than the hot gas column, consequently the two are not in equilibrium. Hence, there is a movement of the cool air to replace the lighter heated gases. If the two columns were brought into equilibrium, it would be necessary to increase the height of the hot column by an additional amount equal to h_c. The existing unbalanced condition promotes the flow of gases within the chimney, and h_c is a measure of the force of the draft.

If equilibrium were established the following relations would exist for volumes of equal cross-sectional areas, and the volumes Q would be proportional to the absolute temperatures of the gases.

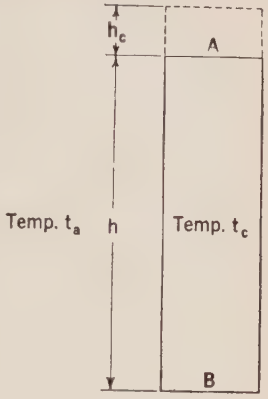


FIG. 106.—Elementary chimney.

$$Q : (Q + Q_c) = h : (h + h_c) = (460 + t_a) : (460 + t_c)$$

$$h(460 + t_c) = (h + h_c)(460 + t_a)$$

$$h(460 + t_c) = h(460 + t_a) + h_c(460 + t_a)$$

$$h(t_c - t_a) = h_c(460 + t_a)$$

$$h_c = \frac{h(t_c - t_a)}{460 + t_a} \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

The unbalanced head h_c is expressed in terms of feet of head of the hot gases so that it is desirable to establish a relation between h_c and the desired draft h_w measured in inches of water. When the density of the flue gases d and the density D of the draft gage fluid are expressed in pounds per cubic foot, h_c may be written

$$h_c = \frac{h_w D}{12d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (101)$$

$$\frac{h_w D}{12d} = \frac{h(t_c - t_a)}{460 + t_a}$$

$$h = \frac{h_w D(460 + t_a)}{12d(t_c - t_a)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (102)$$

in which h = height of chimney, ft.

D = weight of 1 cu. ft. of water at the temperature of the draft gage fluid, lb. Table XIV.

d = density of the flue gases, lb. per cu. ft.
t_c = average temperature of flue gases, deg. fahr.
t_a = temperature outside air, deg. fahr.

The quantity h_w is the theoretical draft in inches of water. Actually, the chimney is not likely to produce this draft, because of leakage, friction and velocity of flow losses. A rough approximation of the actual draft may be 0.80 of h_w.

The cross-sectional area of the chimney is calculated by using the volume of flue gases produced per second and the actual velocity of the gases. The theoretical velocity produced by the unbalanced head h_c may be expressed by a fundamental equation of Mechanics:

$$V = \sqrt{2gh_c}$$

V = theoretical velocity, ft. per sec.
g = acceleration due to gravity, 32.17 ft. per sec. per sec.

$$V = \sqrt{2 \times 32.17}\sqrt{\frac{h(t_c - t_a)}{460 + t_a}} = 8.02\sqrt{\frac{h(t_c - t_a)}{460 + t_a}} \quad . \quad (103)$$

Under actual conditions of chimney operation the flue-gas velocities are only 30 to 50 per cent of the theoretical velocities, due to roughness of the interior surfaces of the chimney, etc. The chimney cross-sectional area is

$$A = \frac{Q}{V \times K} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (104)$$

$$A = \frac{Q}{8.02K\sqrt{\frac{h(t_c - t_a)}{460 + t_a}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (105)$$

in which A = chimney area, sq. ft.
Q = volume of gases handled, cu. ft. per sec.
K = coefficient of velocity, 0.3 to 0.5.
V = theoretical gas velocity, ft. per sec.

Example.—A steam boiler plant consumes 20,000 lb. of coal per hour. The weight of dry flue gases formed per pound of coal fired is 20 lb. The gases enter the chimney at a temperature of 650 deg. fahr. and the average chimney gas temperature is 500 deg. fahr. A chimney is desired that will produce a theoretical draft of 0.9 in. of water at its base when the outside air temperature is 90 deg. fahr. What must be the chimney dimensions?

Solution.—

$$h = \frac{0.9 \times 62.11\ (460 + 90)}{12 \times 0.04138\ (500 - 90)} = 151 \text{ ft.} \quad \text{Ans.}$$

$$Q = \frac{20{,}000 \times 20}{0.03509 \times 60 \times 60} = 3168 \text{ cu. ft. per sec.}$$

$$A = \frac{3168}{8.02 \times 0.4 \sqrt{\frac{151(500 - 90)}{460 + 90}}} = 93.2 \text{ sq. ft.}$$

$$\sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 93.2}{3.1416}} = 10.83 \text{ ft. dia.} \quad \text{Ans.}$$

The densities of flue gases may be taken with small error, as those of dry air, at the same temperature and pressure, and were so used in the foregoing example. Gas or air densities are readily calculated by the use of equation 15, $PV = MRT$. See also Table XVII.

138. Kent's Chimney Formula.—One of the empirical equations frequently used in chimney design is that proposed by William Kent. This formula is based on a coal consumption of 5 lb. per boiler hp. per hr., and is

$$\text{Boiler hp.} = 3.33\ A_e\sqrt{h} \quad . \quad . \quad . \quad . \quad . \quad (106)$$

A_e = effective area of the chimney, sq. ft. An allowance for a dead gas space 2 in. in thickness around the inside of the chimney walls was made by Kent.

$A_e = A - 0.60\sqrt{A}$ where A = the actual chimney area, sq. ft.

h = chimney height, ft.

139. Mechanical Draft.—Draft produced by the action of steam or air jets or centrifugal fans and blowers is termed *mechanical draft*. These methods of production are used where positive and certain draft must be maintained irrespective of weather or boiler-operating conditions and where the natural draft provided by stacks or chimneys is deficient. Boiler auxiliary equipment such as economizers and air preheaters offers resistance to the flow of flue gases as well as reduces the flue-gas temperatures. In such cases a chimney necessary to produce the required amount of draft may be excessive in height and cost, so that mechanical draft is necessary. In general, mechanical draft is desirable, when boilers are to be operated at capacities greatly in excess of their rating or when they must quickly supply sudden demands for steam.

Boilers provided with mechanically produced draft do not require stacks or chimneys as great in height and as expensive in cost as those

necessary for natural draft. In some cases where mechanical draft is used, chimneys of considerable height are built to prevent the smoke and flue gases, together with the cinders and fine ashes that may be discharged, from being objectionable to adjacent property owners. The initial cost, maintenance, depreciation, and operating costs of mechan-

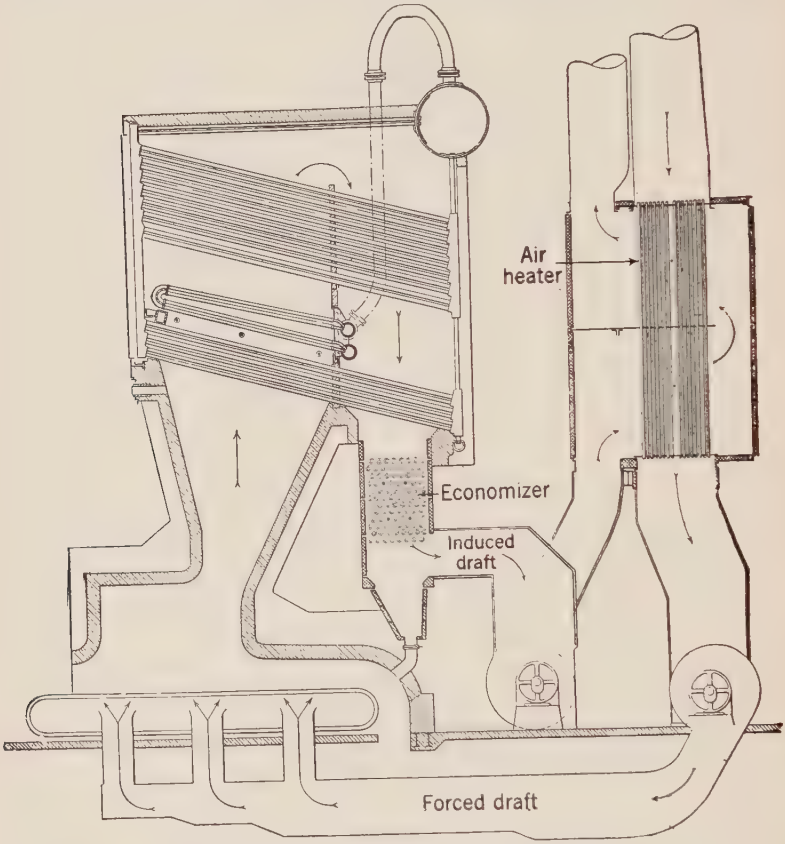


Fig. 107.—Forced- and induced-draft installation.

ical draft equipment must be balanced against similar costs of stacks or chimneys necessary to produce the same amount of draft when a selection is to be made between the two systems.

140. Mechanical Draft Systems.—Two systems are in common use. These are *forced* draft and *induced* draft. Either system may be used alone or in combination as illustrated by Fig. 107.

Forced draft is secured by blowing air into a closed boiler room, into chambers or air boxes under stokers or grates, or through pulverized-fuel

burners. Air is supplied under pressure and thus forces its way through the fuel bed or burner into the furnace. The general practice in forced draft is to remove the products of combustion from the boiler setting by natural or induced draft or a combination of the two. The air supply fans or blowers do not force the products of combustion out of the stack.

Induced draft is procured by a jet blower or a centrifugal fan located in the smoke connection between the boilers and the stack or at the base of the stack. When an economizer is installed the induced draft equipment is placed between the outlet end of the economizer and the stack.

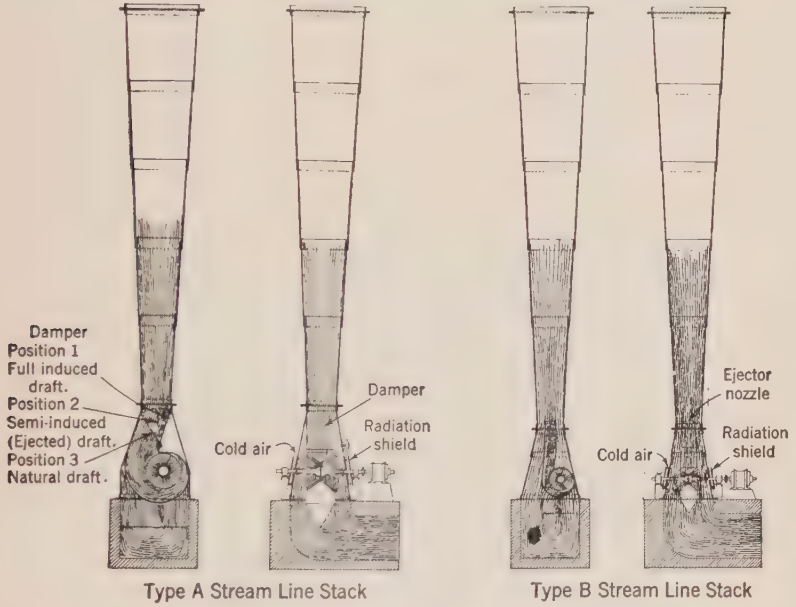


Fig. 108.—Prat-Daniel streamline stacks.

The action of induced-draft apparatus is to reduce the pressure of the gases within the boiler setting below that of the atmosphere. The gases are discharged into the stack under a positive pressure.

Jets of steam discharged through the nozzles of aspirating devices located within the stack near the smoke connection will produce induced draft. Ordinarily the use of such equipment with live steam is not to be recommended as it is not economical in operation.

Induced-draft appliances may produce a pressure within the furnace well below that of the atmosphere so that the outside air pressure is sufficient to force the requisite amount of air through the fuel bed. In the case of underfeed stokers forced draft is required for the air supply.

When a combination of forced and induced draft is used with practically atmospheric pressure in the furnace above the fire, the term *balanced draft* is applied.

141. Evasé Stacks.—The streamline stack for the production of induced draft has been used to a considerable extent in Europe and is finding usage in this country. Two types of these stacks are illustrated by Fig. 108. The type *A* stack is suitable for drafts greater

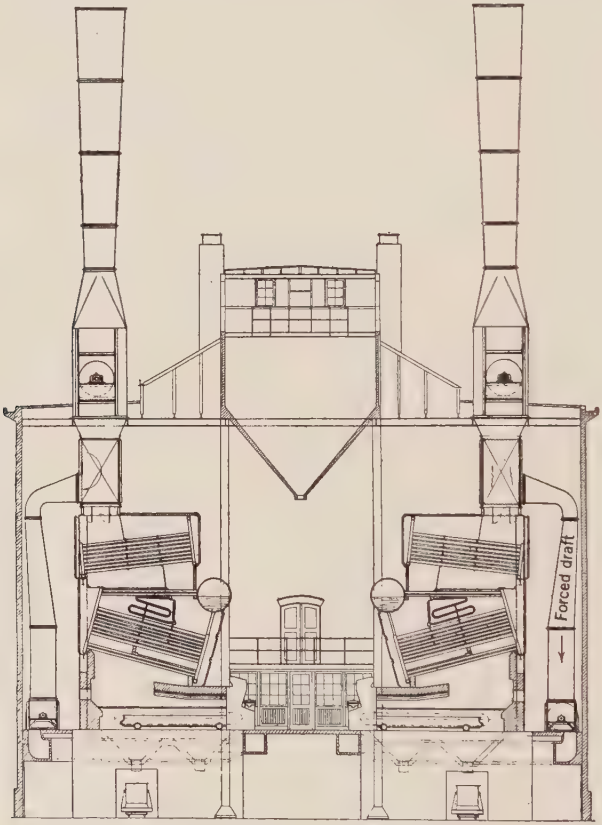


Fig. 109.—An installation of Evasé or streamline stacks.

than 1.5 in. of water and the type *B* stack for drafts less than 1.5 in. of water. Either type of stack may be operated to give full induced draft or natural draft without the fan in operation. Where corrosive or very hot gases are handled, type *B* stacks may have the fan located externally; in this case the fan blows a jet of cool air through the ejector nozzle. Type *A* stacks are built with single and twin fans.

The stack sides form an angle of about 7 deg. with the vertical and

an ejector action is secured as the gases pass at high velocities through the constricted section. The reduction of the velocity of the gases is gradual in the flared part of the structure and a considerable amount of resistance to gas flow is eliminated. A saving in the amount of power required to handle the gases is effected. The stacks are comparatively light in weight and usually have very short smoke pipe connections to the boilers. A typical installation of Evasé stacks with economizers, air preheaters and forced-draft fans is shown by Fig. 109.

142. Centrifugal Fans.—The air- or gas-moving equipment of either forced- or induced-draft systems usually employs centrifugal fans of the following types:

(1) Steel plate
(2) Multiblade
(3) Conoidal
(4) Radial flow
(5) High-pressure blowers

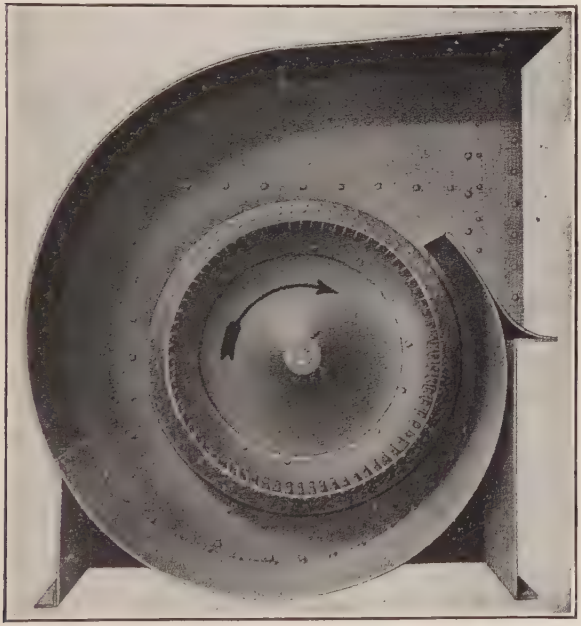


FIG. 110.—New York Blower Company fan wheel and scroll.

All fans consist of a wheel which is rotated within a housing usually constructed of sheet metal. The housing is of a scroll shape, as in Fig. 110, that permits the air or gas to be carried away from the periphery of the wheel with slight disturbances and power losses.

Fan wheels have some sort of vanes or blades located at or near the wheel rim. The action produced by a wheel in rotation is secured as a result of the tendency of air, adjacent to the forward sides of the fan blades, to flow radially outward. This action is due to centrifugal force and the air is discharged from the tips of the blades into the scroll of the housing. As a result of this movement a pressure less than atmosphere is produced at the center of the wheel and a positive air pressure is built up in the scroll. Air or gas flows axially into the wheel to take the place of that discharged. Consequently centrifugal fans may be used not

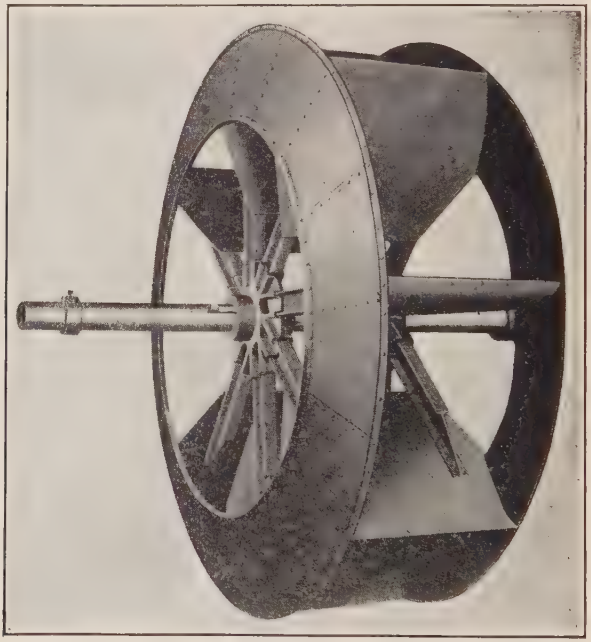


Fig. 111.—American Blower Corporation steel-plate fan wheel.

only to exhaust air or gases through a duct system attached to the fan-housing inlet, but they may also be made to discharge the same air or gases under several inches of water pressure through a duct system leading away from the housing scroll.

Steel Plate Fans.—In fans of this type the wheels consist of one or two spiders, each having from 6 to 12 arms. Each arm or pair of arms carries a flat steel float or blade of some radial length as in the typical wheel of Fig. 111. The wheel floats may be straight or curved either forward or backward depending upon the operating characteristics desired. The blade curvature has a marked effect upon the performance of a given type of fans.

Multiblade Fans.—Such fan wheels are built up of two or more annular rings with many narrow curved blades inserted between them

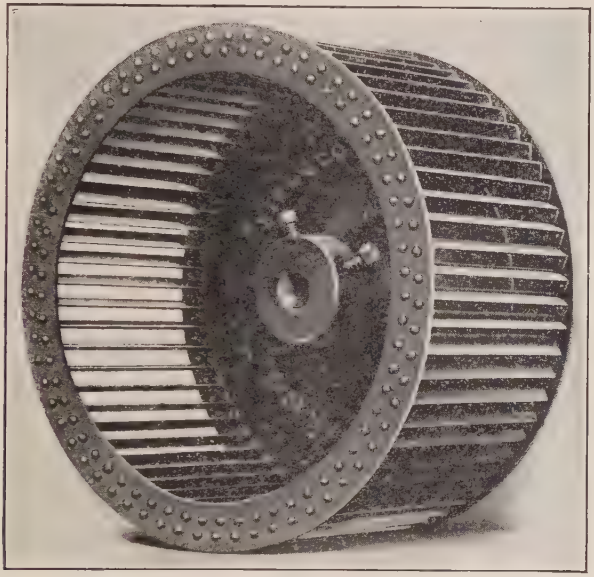


FIG. 112.—Sirocco fan wheel.

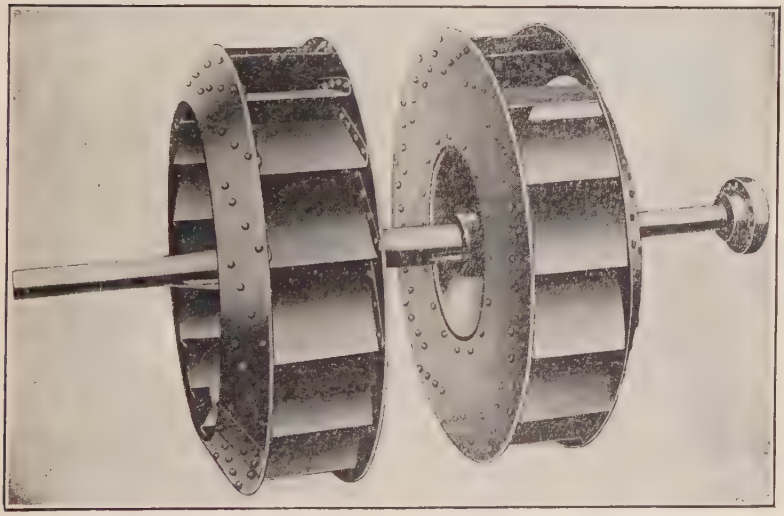


FIG. 113.—American Blower Corporation high-speed fan wheels.

as illustrated by Figs. 112 and 113. The blades may be curved forward or backward depending upon the performance characteristics desired.

The wheel illustrated in Fig. 112 is not adapted to high pressures and high speeds. Multiblade wheels designed for high speeds and pressures are reinforced by a number of annular rings and the blades are made considerably shorter laterally, in order to secure greater rigidity. Fig. 113 illustrates modern high-speed fan wheels with backward curved blades. As shown, two single inlet wheels are mounted upon a shaft used within a double inlet housing. Wheels of this type give high volumetric efficiencies and have a wide range of capacity at constant speed with small changes in the power requirements.

Conoidal Fans.—These fans have wheels as shown by Fig. 114, which are essentially of the multiblade type. However, the term conoidal is

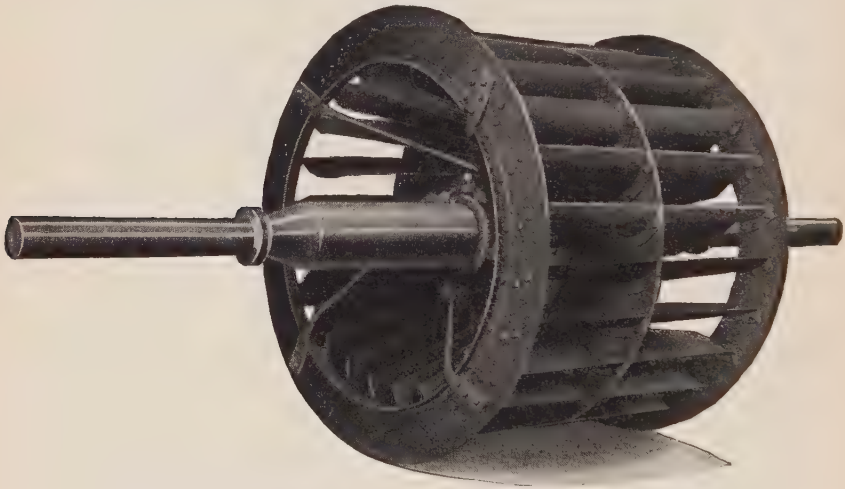


Fig. 114.—Buffalo Niagara conoidal, double-inlet, double-width fan wheel.

applied to blades that are in the form of the surfaces of two tangent cones. This design gives the blade a forward curve at its inward edge and a marked backward curvature at the periphery or exit edge. A blade producing desirable operating characteristics is secured which also has sufficient rigidity without the use of stiffening rings.

Radial Flow Fans.—Wheels of this type have many blades with a backward curvature as shown by Fig. 115. The blade curvature is such that the air leaves the blades quietly with a minimum of shock losses. These fans operate with a characteristic that shows a decrease in the power necessary to drive them with an increase in capacity after a maximum power requirement has been reached.

High-pressure blowers are used only for the production of forced draft. Steel plate fans due to the nature of their design are best suited

for slow rotative speeds and may be directly connected to steam engines so that speed control can be obtained by manipulation of the engine throttle. Steel plate fans may be driven through belt connections from high-speed electric motors when the proper selection of a ratio between driving and driven pulley diameters is made. In such cases variation of the speed may be obtained by use of speed control devices for the motor, but these add materially to the initial cost of the system.

Fans of the conoidal, radial flow, and backward-curved multiblade types are high-speed machines suitable for direct connection to either steam turbines or electric motors. Numerous installations have been made in which fans of these types have been provided with individual direct drives from both steam turbines and motors. Where dual drives are installed, only one source of power can be used at one time. The duplication of the power supply permits the use of the drive most suitable in varying conditions of operation and also insures reliability of fan operation in case of the failure of either the turbine or the motor. In general, the high-speed fans are more efficient than the older and slower speed steel plate fans.

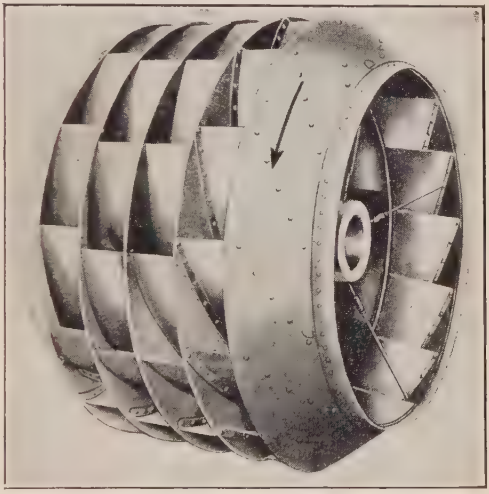


FIG. 115.—Green double-inlet radial-flow fan wheel.

Fans of any of the foregoing classes, when used to produce induced draft or forced draft with preheated air, should be equipped with water-cooled bearings on account of the temperatures of the gases or air handled. Induced-draft fans of all types should be of rugged construction due to the fact that they are subjected to the corrosive action of the flue gases and the abrasion of ash and cinder particles discharged.

The power requirements of induced draft fans handling a given weight of flue gases are greater than those of fans handling the same weight of cool air against the same total pressure in a similar time period.

143. Cindervane Fans.—Modern induced-draft fans operate with high gas velocities and as a consequence considerable quantities of ash and cinder materials are likely to be discharged with the flue gases.

These particles are erosive in their action within the fan and stack and produce a deposit of objectionable debris in the neighborhood of the stack. Fig. 116 illustrates an induced-draft fan designed to eliminate cinder materials from the escaping chimney gases.

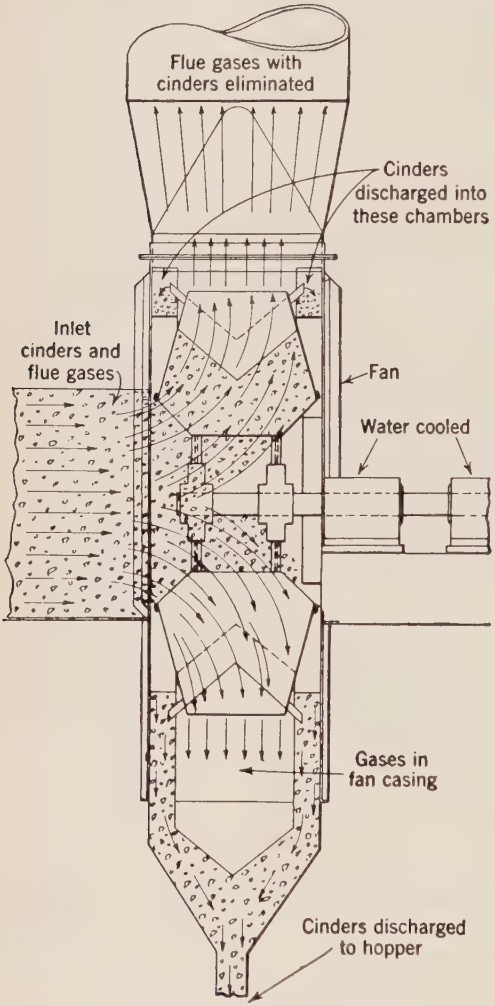


FIG. 116.—Sturtevant cindervane fan.

The separation of the cinder particles is accomplished by the sudden changes of air direction and velocity that occur in all paddle-wheel types of fans. The cinders and flue gases enter the fan in the usual manner at the side of the housing. Attached to the fan blades or floats are steel channels, which catch the cinders and direct them into chambers at the sides of the casing. The space occupied by the fan is only a little greater than that of a standard fan of the same air or gas capacity. Approximately 10 per cent more power is required to operate a cindervane fan than is required for a fan of similar type and capacity without the eliminators.

144. Fan Nomenclature.—Fans of any type have designations applied to them according to the

(1) Number of inlets—single or double.
(2) Width of the wheel, single or double.
(3) Discharge—top, bottom, vertical, horizontal, or angular.
(4) Housing—full, seven-eighths, or three-quarters.
(5) Right or left hand.

A full-housed fan is one in which the fan scroll is completely above the base upon which the fan rests as in the case illustrated by Fig. 110. Seven-eighths or three-quarters full-housed fans have the scroll extending below the top of the supporting base. More of the scroll is located below the base line in the three-quarters full housed than in the case of the seven-eighths full-housed fan. Each builder has his own method of designating fans as right or left hand so that the manufacturer's catalog should be consulted as to the hand of any particular fan.

145. Fan Pressures.—Fans in operation maintain a *total pressure* which is made up of two components, *velocity pressure* and *static pressure*. Under certain conditions of operation either of these two components may be equal to zero, but both are never equal to zero when the fan is running.

Velocity pressure is utilized in creating and maintaining the velocity of air or gas flow.

Static pressure is one that tends to burst the housing or duct systems along either longitudinal or girth seams. Static pressure is utilized in overcoming frictional and other resistances offered to the flow of air or gases. Static pressure, velocity pressure, and total pressure are all inter-related. Thus, if the gas handled has a velocity increase in the duct system, a part of the static pressure available at that particular locality is used to create the additional velocity pressure necessary. Likewise, if the speed of flow is reduced at some point a part of the velocity pressure at that point will be converted back into static pressure.

Any fan which develops a large static pressure in comparison with its total pressure has a greater capacity to overcome the resistances offered to air or gas flow, by duct friction, boiler-setting baffles, or fuel bed obstructions. In the case of steel plate fans the most economical operation is secured when the ratio of the total to the static pressure is between 1.2 and 1.3, while for multiblade fans the most satisfactory ratio is between 1.05 and 1.15.

146. Measurement of Fan Pressures.—As fans create pressures less than atmospheric on their suction side, and produce pressures greater than atmospheric on the discharge side, consideration must be given to pressure measurements in both the inlet and outlet ducts. The pressure measurements are made somewhat differently in the two cases.

Pressures greater than atmospheric.—Fig. 117 illustrates a duct of circular section in which air is flowing at some velocity with a static pressure above that of the air outside the duct. The static pressure existing within the duct at any location may be obtained at that point by attaching a draft gage or manometer of sufficient capacity to a con-

nection made at right angles to the longitudinal axis of the duct as at A, Fig. 117. The gage connection should not extend through the duct wall and the small opening should be free from burrs or irregularities. Due to the fact that turbulence of flow may exist in the duct, it is likely that the static pressure will vary at different points about the circumference of the duct section. Therefore, it is desirable that several static pressure openings in the duct at the plane of the section be joined by a piezometer ring, Fig. 118, to the manometer. With such an arrangement the average static pressure may be more accurately secured.

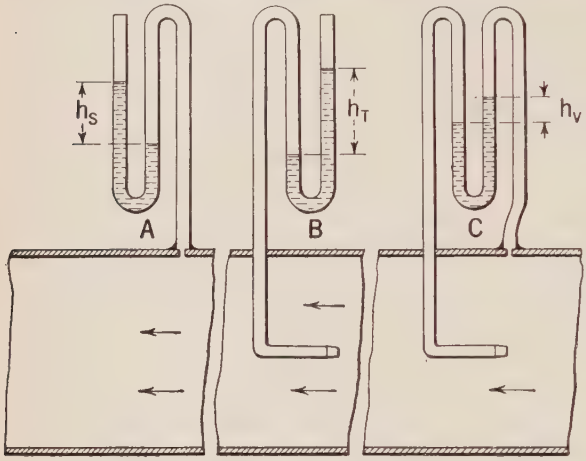


FIG. 117.—Air pressure measurements,—greater than atmospheric pressure.

The measurement of velocity pressure is accomplished by the determination of the total pressure and the subtraction of the static pressure from it, either mathematically or by the proper attachment of the static and total pressure connections to the indicating gage. If a piece of tubing of small internal diameter and thin walls is bent at right angles and has the end of the short leg cut squarely across and the tube walls ground to a thin edge, a device known as a Pitot tube is formed. The tube is inserted into the duct at B, Fig. 117, with the bent portion of the tube located parallel to the longitudinal axis of the duct, so that the moving air will flow directly against the open end of the tube. When the outer end of the tube is attached to a manometer the impact of the

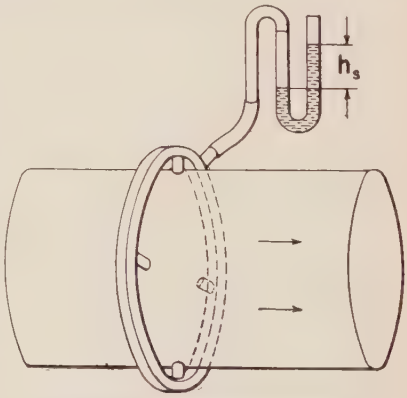


FIG. 118.—Piezometer ring.

air as well as the static pressure at that particular location will be indicated. The reading so obtained is the total pressure. When the velocity pressure alone is desired the total and static pressure attachments are made to a single gage as at *C*, Fig. 117.

A commercial type of Pitot tube as adopted by the American Society of Heating and Ventilating Engineers, Fig. 119, combines in a single instrument both static and total pressure openings. Pitot tubes are not adapted to the measurement of velocity pressures corresponding to gas flows of less than 6 lin. ft. per second unless used with very sensitive and delicate pressure-measuring gages.

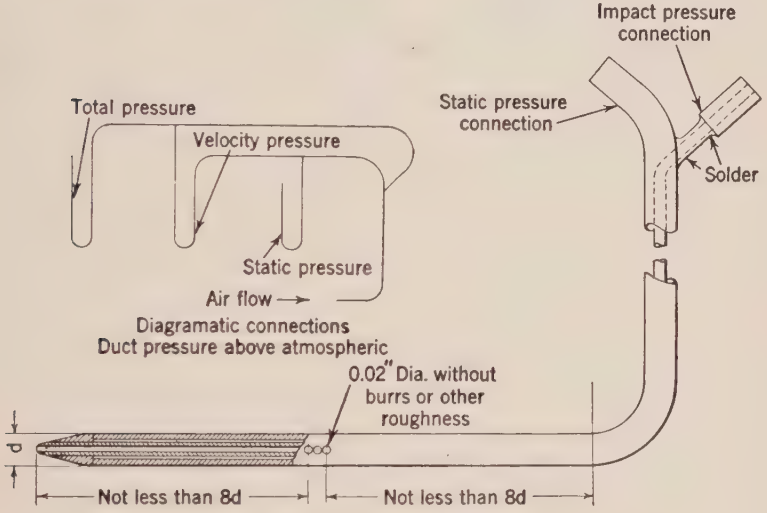


Fig. 119.—Pitot tube.

Velocity-pressure measurements should be made in a straight section of pipe at least 20 duct diameters in length, with at least 10 duct diameters of straight pipe on either side of the Pitot tube station. Erroneous or variable data may be secured when the observations are taken near a fan outlet or a bend in the duct.

147. Pressures Less than Atmospheric.—When the pressure within a duct is less than atmospheric, as on the suction side of a fan, Fig. 120, the measurements of static, total, and velocity pressures are not indicated as they are for pressures greater than atmospheric.

Due to the fact that the fan produces a partial vacuum at its wheel center the absolute pressure within the duct is reduced to below that of the absolute pressure of the air outside of the duct. The difference in absolute pressure is equal to the total pressure difference produced on the

suction side of the fan. This total pressure is the sum of the velocity pressure necessary to create the actual velocity of flow and the static pressure maintained to overcome friction and other losses in the suction duct. The manometer connection shown at *A*, Fig. 120, measures the difference between the absolute pressures within and without the duct and thus indicates the total pressure and not the static pressure alone as for pressures greater than atmospheric. When a Pitot tube is connected to a gage as at *B*, Fig. 120, the impact or velocity pressure partially neutralizes the total pressure difference and the static pressure is obtained. If the velocity pressure alone is desired the usual connections of a Pitot tube for velocity pressure are made as at *C* in Fig. 120.

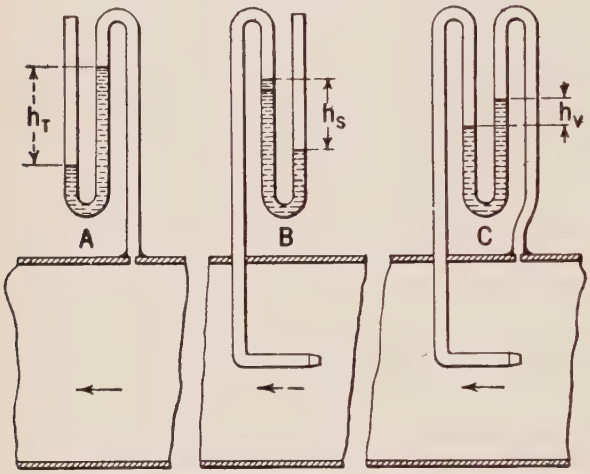


FIG. 120.—Air pressure measurements,—less than atmospheric pressure.

148. Velocity of Flow.—The velocities of gas flow may be readily computed from measured velocity pressures by the equation

$$V = \sqrt{2gh}$$

in which V = velocity in feet per second.
g = acceleration due to gravity, 32.17 ft. per sec. per sec.
h = distance in feet through which a falling body acquires a velocity of V feet per second.

As the factor h is expressed in feet, it is necessary to convert the measured velocity pressure into feet head of the fluid flowing as

$$h = \frac{Dh_v}{12d}$$

in which h = head of the fluid flowing, ft.
h_v = velocity pressure measured, inches of water.
D = weight of 1 cu. ft. of water at the temperature of the gage fluid. Table XIV.
d = density of the fluid flowing, lb. per cu. ft.

The foregoing equations may be combined into

$$V = \sqrt{2g\frac{Dh_v}{12d}}$$

$$= \sqrt{2g}\sqrt{\frac{D}{12d}}\sqrt{h_v} \quad . \quad . \quad . \quad . \quad . \quad . \quad (107)$$

Example.—Find the velocity of flow corresponding to a velocity pressure of 0.25 in. of water when air at 70 deg. fahr. is being handled. Barometer, 29.92 in. Hg. Temperature of the gage fluid, 70 deg. fahr.

Solution.—

$$V = \sqrt{2 \times 32.17}\sqrt{\frac{62.3}{12 \times 0.075}}\sqrt{0.25} = 33.4 \text{ ft. per sec.}$$

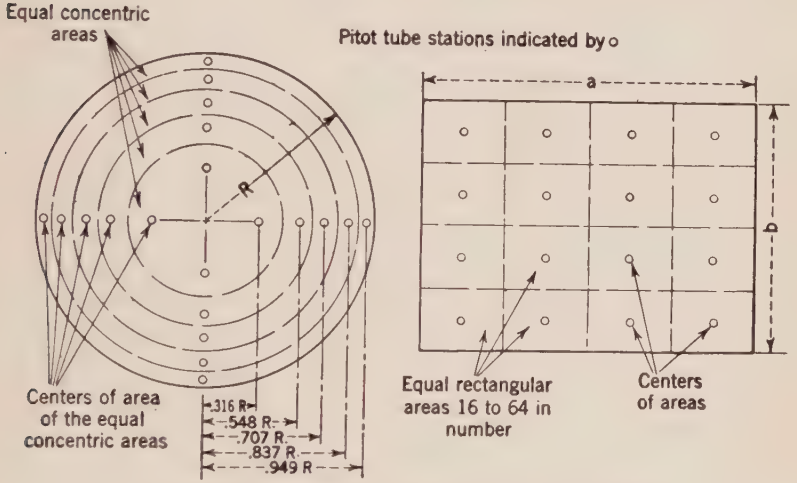


Fig. 121.—Locations of a Pitot tube for a duct traverse.

The velocity pressure measured by the use of a Pitot tube is the one existing at the open end of the tube. The velocity of flow is not uniform over a duct cross-section so that several velocity-pressure determinations are required to secure the average velocity of flow existing at any Pitot tube station. The duct cross-section is usually divided into a number of equal areas and observations are made with the tube properly located within these areas. A scheme of locating a tube at various points on two diameters of a circular duct cross-section and in a rectangular or square duct is illustrated by Fig. 121. Velocity-pressure observations made at points on diameters that are at right angles to each other are used for the calculation of the average velocity of flow in the circular duct. The square root of the velocity pressure is used in equation

107; therefore, the average square roots of the velocity pressures taken should be computed and not the average of the velocity pressures read. Thus

$$h_{v_a}^{1/2} = \frac{\sqrt{h_{v_1}} + \sqrt{h_{v_2}} + \sqrt{h_{v_3}} \ldots + \sqrt{h_{v_n}}}{n} \quad . \quad . \quad . \quad (108)$$

where n is equal to the number of observations taken.

Knowledge of the relation of the average velocity in the duct to the velocity existing at the duct center is desirable. The center velocity is usually but not always greater than the average velocity of flow. The ratio is known as the *center coefficient* and is expressed as

$$C = \frac{V_a}{V_c} = \frac{\sqrt{h_{v_a}}}{\sqrt{h_{v_c}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (109)$$

where C = center coefficient.
V_a = average velocity in duct, ft. per sec.
V_c = velocity at the center of the duct, ft. per sec.

The center coefficient varies with the velocity of flow, the density of the gas handled and the shape of the duct cross-section. Usual values of C for ducts of circular shape range from 0.91 to 0.93, but ducts of other shapes may have coefficients greater or less than these values. When the center coefficient C has been accurately determined the center velocity pressure h_{vc} only need be measured and the average duct velocity may be expressed as

$$V_a = C\sqrt{2g}\sqrt{\frac{D}{12d}}\sqrt{h_{v_c}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (110)$$

149. Volume of Discharge.—The volume of gas flowing is equal to the product of the duct section in square feet and the average velocity of flow at that section.

$$Q = AV_a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (111)$$

in which Q = cubic feet discharged per sec.
A = duct cross-section area, sq. ft.
V_a = average velocity of flow, ft. per sec.

150. Total Pressure Developed by a Fan.—The total pressure exerted is the sum of the maximum average velocity pressure and the discharge static pressure, minus the static pressure in the inlet duct. When the pressure in the inlet duct is less than atmospheric, the ob-

served static pressure is given a minus sign, so that the inlet static pressure is actually added when the algebraic solution is made. When the inlet duct pressure is greater than atmospheric, the inlet static pressure is positive and is actually subtracted.

Total pressure = (maximum average velocity pressure)
+ (discharge static pressure) − (suction static pressure) . (112)

Fan suction inlets are commonly made larger than the discharge outlets so that the maximum velocity pressure is generally at the fan outlet. When the quantity of air being discharged is known as well as the area of the fan outlet, the velocity of the air leaving the fan casing and the corresponding velocity pressure can be computed and used in equation 112.

151. Fan Air Horsepower.—This power output represents the work done by the fan and is

$$\text{air horsepower, } a.hp. = \frac{W_a H}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad (113)$$

W_a = weight of air handled per minute, lb.
H = total head against which the fan works, feet head of air.

In terms of the total fan pressure

$$H = \frac{h_t \times D}{12d}$$

where h_t = total pressure exerted by the fan, in. of water.
D = density of 1 cu. ft. of water at the temperature of the gage fluid, lb.
d = density of the gas handled, lb. per cu. ft.

$$a.hp. = \frac{W_a \times h_t \times D}{12 \times d \times 33{,}000} \quad . \quad . \quad . \quad . \quad . \quad (114)$$

$$W_a = Q \times d \times 60$$

$$a.hp. = \frac{Q \times 60 \times h_t \times D}{12 \times 33{,}000} \quad . \quad . \quad . \quad . \quad (115)$$

Example.—A fan discharges 24,000 cu. ft. of air per minute through its outlet of 12.6 sq. ft. area and maintains a static pressure of 5 in. of water. Air temperature, 70 deg. fahr. Barometer, 29.92 in. Hg. Find the air horsepower developed.

Solution.—

$$V_a = \frac{24{,}000}{12.6 \times 60} = 31.8 \text{ ft. per sec.}$$

$$h_{v_a} = \frac{12 V_a^2 d}{2gD} = \frac{\overline{31.8}^2 \times 12 \times 0.075}{2 \times 32.17 \times 62.3} = 0.226 \text{ in. of water}$$

$$h_t = h_s + h_{v_a} = 5. + 0.226 = 5.226 \text{ in. of water,}$$

$$a.hp. = \frac{24{,}000 \times 5.226 \times 62.3}{12 \times 33{,}000} = 19.8. \quad Ans.$$

152. Mechanical Efficiency of Fans.—The ratio of the air horsepower output to the driving power required at the fan shaft expresses the mechanical efficiency of the fan.

$$\text{Fan mechanical efficiency} = \frac{\text{air horsepower}}{\text{horsepower input}} \quad . \quad . \quad (116)$$

Example.—In the preceding example 26.8 brake horsepower were required to drive the fan. What was the mechanical efficiency?

Solution.—

$$\text{Efficiency} = \frac{19.8 \times 100}{26.8} = 73.9 \text{ per cent.}$$

The mechanical efficiencies of steel plate fans range from 40 to 60 per cent and those of multibladed fans from 50 to 80 per cent. Each fan has a condition of maximum efficiency when working against any static pressure. Fan builders usually signify, by bold face type in their capacity tables, the speed and brake horsepower necessary to give the maximum efficiency at various static pressures.

153. Fan Performance.—Fan characteristics and performance are dependent upon the design and conditions of operation, that is, speed, temperature of the gas handled, static pressure, and the velocity pressure maintained. Table X gives the performance data of multiblade fans suitable for forced draft. These numerical quantities were selected from the builder's data as those representing a condition of maximum efficiency at the various static pressures. The individual performances of a single fan selected from the data of Table X are shown graphically by the curves of Fig. 122.

Fan capacity varies directly as the speed ratio, the static pressure developed as the square of the speed ratio, and the power required to drive the fan as the cube of the speed ratio. Thus doubling the speed of a fan doubles the volume of its discharge, produces a static pressure four times as great, and requires eight times as much power to drive the

TABLE X

CAPACITIES, SPEEDS, AND HORSEPOWERS, GREEN RADIAL-FLOW DOUBLE-INLET, FORCED-DRAFT FANS

(Green Fuel Economizer Co.)

Air temperature, 70 deg. fahr. Barometer, 29.92 in Hg.
Table values are for maximum efficiency

Fan No.	Wheel Dia., In.	Performance	Static Pressure at Fan Discharge, Inches of Water						
			1	2	3	4	5	6	7
0	20½	C.f.m.	2,800	4,000	5,200	6,000	6,400	7,200	7,600
		R.p.m.	860	1,216	1,505	1,738	1,927	2,122	2,280
		B.hp.	0.62	1.77	3.49	5.36	7.15	9.65	11.85
1	24	C.f.m.	4,000	5,500	7,000	8,000	9,000	9,500	10,500
		R.p.m.	741	1,043	1,284	1,481	1,658	1,805	1,957
		B.hp.	0.87	2.47	4.69	7.15	10.01	12.8	16.4
2	27¾	C.f.m.	5,600	7,700	9,100	10,500	11,900	12,600	14,000
		R.p.m.	648	913	1,110	1,280	1,435	1,563	1,696
		B.hp.	1.22	3.41	6.11	9.39	13.4	17.0	21.9
3	32¼	C.f.m.	7,200	9,900	12,600	14,400	16,200	17,100	18,900
		R.p.m.	550	776	955	1,102	1,232	1,342	1,456
		B.hp.	1.57	4.41	8.46	12.9	18.1	22.9	29.5
4	37¼	C.f.m.	10,000	13,750	16,250	18,750	21,250	23,750	25,000
		R.p.m.	478	671	816	944	1,055	1,161	1,248
		B.hp.	2.18	6.13	10.9	16.8	23.8	31.8	39.0
5	43	C.f.m.	13,500	18,000	22,500	25,500	28,500	31,500	33,000
		R.p.m.	416	583	714	821	917	1,007	1,083
		B.hp.	2.97	8.06	15.1	22.8	31.9	42.3	51.6
6	50	C.f.m.	18,000	24,000	30,000	36,000	38,000	42,000	46,000
		R.p.m.	354	495	609	707	782	859	930
		B.hp.	3.95	10.7	20.1	32.3	42.5	56.3	71.9
7	58	C.f.m.	22,000	34,000	40,000	46,000	52,000	58,000	61,000
		R.p.m.	301	433	525	606	678	746	803
		B.hp.	4.86	15.2	26.9	41.3	58.0	77.8	95.9
8	67¼	C.f.m.	30,000	46,000	54,000	62,000	70,000	78,000	82,000
		R.p.m.	260	372	452	521	583	641	689
		B.hp.	6.57	20.5	36.1	55.6	78.1	104	128
9	78	C.f.m.	45,000	60,000	75,000	85,000	95,000	105,000	110,000
		R.p.m.	228	319	391	451	504	553	594
		B.hp.	10.2	26.8	50.2	76.0	107	141	172
10	90½	C.f.m.	57,500	80,000	102,500	117,500	125,000	140,000	155,000
		R.p.m.	194	274	337	389	431	474	515
		B.hp.	12.7	35.5	68.5	105	140	187	242

fan. An examination of the data of Table X and Fig. 122 will verify the foregoing statement.

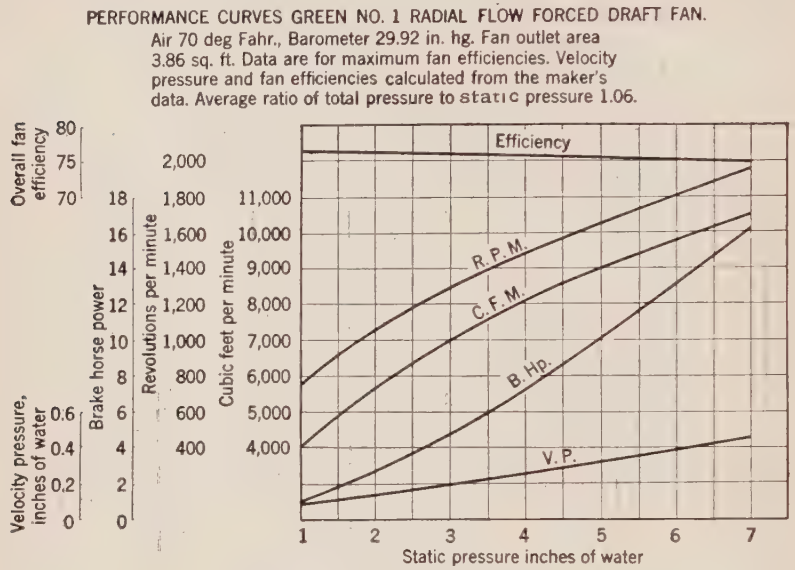

PERFORMANCE CURVES GREEN NO. 1 RADIAL FLOW FORCED DRAFT FAN.
Air 70 deg Fahr., Barometer 29.92 in. hg. Fan outlet area 3.86 sq. ft. Data are for maximum fan efficiencies. Velocity pressure and fan efficiencies calculated from the maker's data. Average ratio of total pressure to static pressure 1.06.
Overall fan efficiency
Brake horse power
Revolutions per minute
Cubic feet per minute
Velocity pressure, inches of water
Efficiency
R.P.M.
C.F.M.
B.Hp.
V.P.
Static pressure inches of water

Fig. 122.—Fan performance curves.

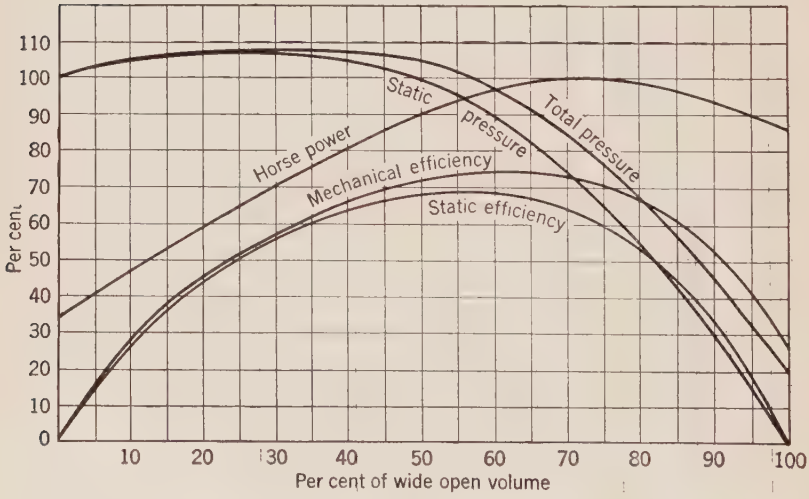


Fig. 123.—Characteristic curves, type RB Green radial-flow forced-draft fans.

Example.—A No. 1 fan at 741 r.p.m. delivers 4000 c.f.m. at 1-in. static pressure and requires 0.87 b.hp. to drive it. When the speed is doubled, or

TABLE XI

Variable Capacities of a No. 1 Green Radial-Flow Forced-Draft Fan

(Green Fuel Economizer Co.)

Capacity, C.f.m.	Static Pressure, Inches of Water Fan Discharge.													
	1		2		3		4		5		6		7	
	R.p.m.	B.hp.	R.p.m.	B.hp.	R.p.m	B.hp.	R.p.m.	B.hp.	R.p.m.	B.hp.	R.p.m.	B.hp.	R.p.m.	B.hp.
2,000	710	.58												
2,500	712	.61												
3,000	718	.68	1004	1.65										
3,500	724	.78	1007	1.77	1230	2.95								
4,000	**741**	**.87**	1012	1.89	1234	3.09	1420	4.50						
4,500	763	1.02	1018	2.06	1238	3.29	1422	4.72	1589	6.28				
5,000	794	1.12	1030	2.25	1242	3.53	1425	4.98	1591	6.57	1741	8.33		
5,500	830	1.31	**1043**	**2.47**	1248	3.80	1430	5.32	1593	6.88	1743	8.69	1880	10.6
6,000	868	1.50	1062	2.70	1256	4.06	1436	5.60	1596	7.25	1745	9.06	1882	11.0
6,500	915	1.69	1089	2.95	1269	4.40	1442	5.96	1600	7.66	1750	9.47	1886	11.5
7,000	958	1.95	1119	3.23	**1284**	**4.69**	1453	6.32	1608	8.07	1755	9.95	1889	12.0
7,500	1005	2.18	1152	3.50	1306	5.02	1465	6.72	1616	8.50	1760	10.4	1895	12.5
8,000	1048	2.49	1190	3.87	1334	5.44	**1481**	**7.15**	1627	8.98	1768	11.0	1901	13.1
8,500	1097	2.80	1230	4.25	1363	5.87	1498	7.61	1640	9.52	1778	11.5	1907	13.6
9,000	1144	3.19	1270	4.69	1397	6.33	1525	8.12	**1658**	**10.01**	1790	12.1	1916	14.3
9,500	1194	3.58	1320	5.17	1435	6.81	1556	8.64	1679	10.7	**1805**	**12.8**	1928	15.0
10,000	1242	3.99	1360	5.63	1475	7.34	1588	9.28	1702	11.3	1821	13.4	1941	15.7
10,500	1292	4.47	1408	6.16	1517	7.92	1624	9.9	1721	12.0	1843	14.2	**1957**	**16.4**
11,000	1342	4.95	1455	6.69	1558	8.55	1660	10.6	1763	12.7	1867	14.9	1974	17.2
11,500	.....	.....	1500	7.34	1603	9.26	1700	11.3	1797	13.5	1896	15.7	1996	18.1
12,000	.....	.....	1547	7.97	1645	9.98	1740	12.1	1840	14.3	1928	16.6	2024	19.0
12,500	.....	.....	1596	8.70	1690	10.7	1783	12.9	1872	15.2	1959	17.5	2055	19.9
13,000	.....	.....	1645	9.42	1735	11.6	1826	13.7	1911	16.1	1997	18.5	2087	21.0
13,500	.....	.....	1695	10.3	1783	12.5	1870	14.7	1952	17.0	2034	19.5	2118	22.2
14,000	.....	.....	1743	11.1	1830	13.4	1915	15.6	1995	18.1	2075	20.6	2156	23.2
14,500	.....	.....	1791	12.0	1875	14.3	1960	16.7	2039	19.1	2115	21.7	2191	24.4
15,000	.....	.....	1840	12.9	1924	15.3	2006	17.8	2082	20.2	2157	22.8	2230	25.6
15,500	.....	.....	1892	13.9	1973	16.4	2052	19.0	2127	21.5	2200	24.2	2271	26.9
16,000	.....	.....	1940	14.9	2020	17.5	2098	20.1	2171	22.7	2241	25.5	2313	28.2
16,500	.....	.....	.....	.....	2070	18.7	2143	21.4	2216	24.1	2286	26.8	2356	29.6
17,000	.....	.....	.....	.....	2120	19.9	2192	22.7	2264	25.5	2330	28.3	2397	31.1
17,500	.....	.....	.....	.....	2170	21.2	2242	24.1	2310	26.9	2378	29.8	2441	32.6
18,000	.....	.....	.....	.....	2216	22.5	2288	25.5	2355	28.4	2420	31.4	2485	34.2
18,500	.....	.....	.....	.....	2267	24.0	2339	27.2	2402	30.0	2468	33.0		
19,000	.....	.....	.....	.....	2319	25.6	2388	28.8	2452	31.7	2513	34.9		

Notes.—C.f.m. = cubic feet of air per minute at 70 deg. fahr. and 29.92 in. Hg. Bold face figures indicate the point of maximum efficiency.

1481 r.p.m., the discharge is 8000 c.f.m., the static pressure 4 in. of water, and the b.hp. 7.15 or 8.2 times as much as at the 741 r.p.m. speed.

Fans may be operated to give variable capacities at any static pressure. Data for such operations are given in Table XI for a No. 1 Green radial-flow forced-draft fan.

With the aid of a set of curves as shown in Fig. 123, the performances of any of the fans listed in Table X may be ascertained when the operation is at various capacities. The static efficiency is the mechanical efficiency multiplied by the ratio of the static to the total pressure.

PROBLEMS

1. Find the theoretical draft at the base of a chimney 125 ft. high when the outside air temperature is 95 deg. fahr. and the average temperature of the chimney gases is 600 deg. fahr.

2. What would be the force of the draft of the chimney of problem 1 if the outside air temperature were −20 deg. fahr.?

3. Find the theoretical draft of a chimney 250 ft. high when the average flue gas temperature is 600 deg. fahr., and the outside air temperature is 95 deg. fahr.

4. Find the velocity of flow of the flue gases for the chimney of problem 1. The actual velocity of flow is assumed to be one-third of the theoretical.

5. A boiler plant requires a theoretical draft of 1.25 in. of water. The coal consumption is 18,000 lb. of coal per hour, and 15 lb. of dry flue gases are formed per lb. of coal fired. Temperature of the gases in the breeching 700 deg. fahr., average temperature of the chimney gases 550 deg. fahr., and of the outside air 80 deg. fahr. Find the chimney dimensions.

6. A chimney is 175 ft. high and 12 ft. inside diameter at the base. Find the chimney capacity in boiler horsepower. Use Kent's formula.

7. Air at a room temperature of 70 deg. fahr. is flowing in a duct with an average velocity of 20 ft. per sec. Barometer, 29.6 in. Hg. Calculate the average velocity pressure. If the static pressure is 0.5 in. of water, what is the total pressure?

8. Air at 80 deg. fahr. flows in a duct 4 ft. in diameter. Velocity pressure at the center of the duct 0.25 in. of water, room temperature 70 deg. fahr., barometer 29.4 in. of Hg, center coefficient 0.92. Find the cubic feet of air handled per minute.

9. A fan blowing air into a duct 5 ft. in diameter has a discharge outlet area equal to that of the duct. Static pressures of 4 in. of water are maintained at the fan outlet. and −0.25 in. of water at the fan inlet. Velocity pressure at the duct center 0.16 in. of water, air temperature 40 deg. fahr., room temperature 50 deg. fahr., barometer 29.3 in. Hg, duct center constant 0.91. Find the air horsepower developed.

10. If the fan efficiency of problem 9 is 60 per cent, what horsepower is required to drive the fan? If the speed of the fan is made 50 per cent greater, what will be the capacity, the static pressure, and the horsepower necessary to drive the fan?

CHAPTER XI

THE STEAM ENGINE

154. Application.—As a source of power the steam engine has the following characteristics as a prime mover:

(1) Ability to start under load.
(2) Nearly constant, slow, or comparatively slow rotative speeds.
(3) Variable rotative speeds.
(4) Direction of rotation may be reversed.

Not every steam engine can satisfy all of the foregoing items, but by proper design all of those mentioned may be incorporated in a single machine as in the modern steam locomotive.

The steam engine is not suited to extremely high rotative speeds and in condensing practice is limited to operation with a maximum vacuum of about 25 to 26 in. of mercury in the exhaust piping. In general, steam engine units of good design up to 2000 hp. in size are as economical as or more economical than steam turbines of similar size operating with comparable conditions. For the production of large quantities of electrical energy the steam engine has been generally superseded by the steam turbine. High rotative speeds are desirable in alternating current generation. The steam turbine can develop a large amount of power in a machine occupying a very much smaller floor space than would be required for steam engines of equal capacity. The turbine can operate with high initial steam pressures and temperatures as well as with vacua of 28 to 29 in. of mercury in the exhaust connections. The latter conditions secure for the turbine an economy of operation that is not equaled by the steam engine.

Steam engines are largely used for driving pumps, fans, blowers, line shafts, rolling mill machinery, hoisting equipment, electrical generators of limited sizes, refrigerating machinery, and air compressors.

155. The Simple Engine.—The simple engine generally has only a single steam cylinder although twin cylinders may be used. In the simple engine partially expanded steam is not discharged from one cylinder to another for further expansion of the steam. Fig. 124 illustrates a small vertical steam engine with a throttling governor and a D slide

valve. Such engines are suitable for the development of small amounts of power and are used for driving fans, blowers, stokers, small electric generators, etc. The essential parts of the engine (Fig. 124) are illustrated and designated by Fig. 125. The engine is double acting because a power cycle is produced in each end of the cylinder during one revolution of the crankshaft.

Steam from the supply line is admitted by the throttle valve and passes through the governor to the steam chest. The D-slide-valve alternately admits steam to and releases the steam from each end of the cylinder. The valve has a reciprocating motion produced by the gear composed of the eccentric, the eccentric straps, the eccentric rod, the valve slide and the valve rod. As shown by Fig. 126, the valve is in a position to admit steam to the upper end of the cylinder and to allow the escape of used steam from the lower end. The piston is moved by the pressure of the steam admitted to the cylinder. At some point of the piston stroke the valve completely closes the supply port and the flow of steam from the steam chest is cut off. After cut-off has taken place the pressure within the cylinder decreases due to expansion of the steam. Energy stored in the heavy flywheel during the period of steam admission is used to maintain the engine speed. Near the end of the piston stroke the valve moves to uncover the steam port of the cylinder end under consideration. When this movement takes place the expanded steam is released and flows to the exhaust passage underneath the valve as shown for the lower (crank) end of the cylinder, Fig. 126. The exhaust of expanded steam underneath the valve occurs while live steam is being admitted to the opposite (head) end of the cylinder and continues until near the end of the return stroke of the piston. When the valve has moved to completely close both ports of the cylinder, some

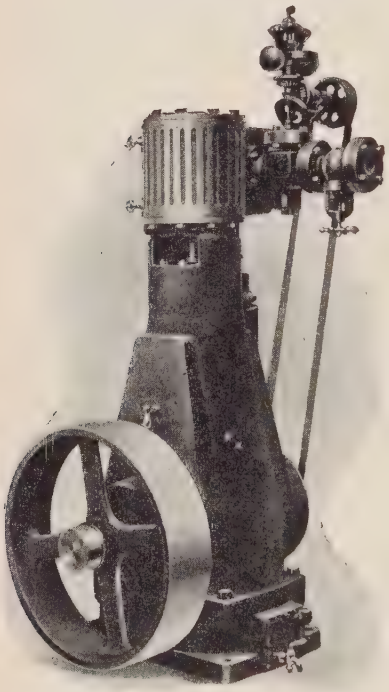


FIG. 124.—Troy vertical steam engine.

expanded steam is entrapped ahead of the moving piston and is compressed with a rise in pressure. This compression acts as a cushion for

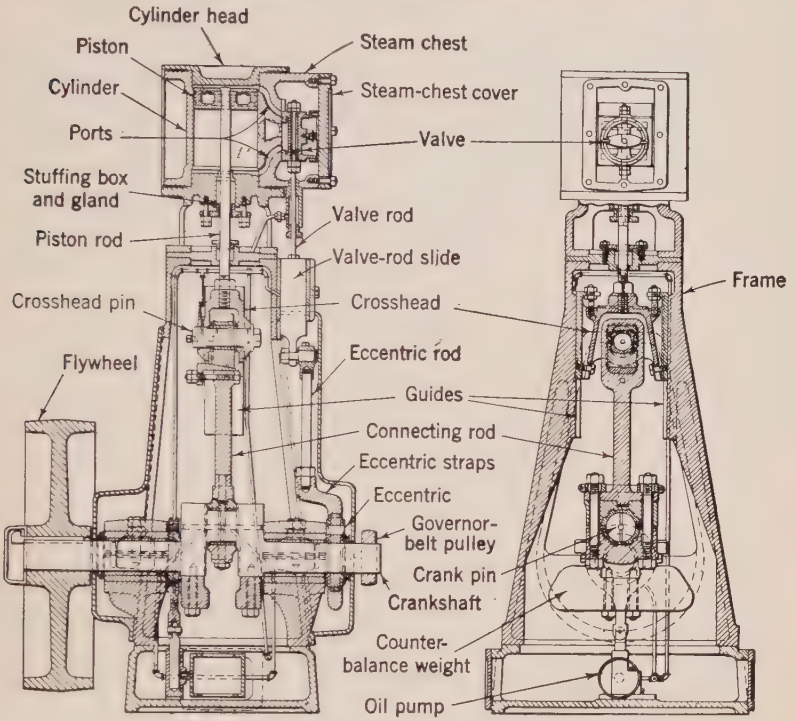


FIG. 125.—Sectional-elevation of a simple vertical steam engine.

the returning piston and continues until the valve has moved to admit live steam to that end of the cylinder. One such complete cycle takes place in each end of the cylinder per revolution.

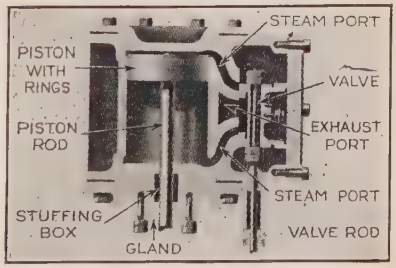


FIG. 126. Section of a simple steam engine cylinder.

The reciprocating motion imparted to the piston by the alternate admission, expansion and release of steam in each end of the cylinder is transmitted by the piston rod to the crosshead. This part moves back and forth between the crosshead guides and delivers its motion to one end of the connecting rod. The connecting rod is fitted at each end with a pin joint and converts the reciprocating motion of the crosshead to rotary motion at the crankshaft. The pin passing through the cross-

head and one end of the connecting rod is known as the crosshead or wrist pin. The pin at the crankshaft end of the connecting rod is called the crank pin.

156. Engine Classifications.—Steam engines are divided into a number of types, depending upon their design, construction, and method of operation. The common classifications used are:

(1) Position of the frame and cylinder { Horizontal / Vertical

(2) Valve gear { Slide valve { D / Balanced / Multi-ported / Piston ; Corliss { Releasing / Non-releasing ; Poppet

(3) Number of cylinders used for steam expansion { Single cylinder, simple / Two cylinders, compound / Three cylinders, triple expansion / Four cylinders, quadruple expansion

(4) Rotative speed { Low / Medium / High

(5) Ratio of stroke to diameter { Long / Short

(6) Exhaust disposal { Condensing / Non-condensing

157. Engine Nomenclature.—In steam engine parlance the following terms are used: *Head end* of the cylinder is farther from, and the *crank-end* is nearer to the crankshaft. *Dead center* is the position at which reversal of piston motion occurs at either end of the cylinder. In the dead center position the longitudinal center lines of the crank, the connecting rod and the piston rod are all in perfect alignment. An engine has two dead centers, that is, head-end and crank-end. *Stroke* is the linear distance traveled by the piston from one dead center position to the other. The forward stroke is made between the head-end and the crank-end dead centers and the return stroke is completed with motion in the opposite direction. Long-stroke engines have a piston travel greater than, while short-stroke engines have piston travels equal to or

less than the cylinder diameter. *Piston displacement* is the volume swept through by the piston during a stroke and is equal to the product of the piston area and the stroke length.

Clearance is either mechanical or volumetric. Mechanical clearance is the smallest linear distance between the cylinder head and the piston at the dead center position. Volumetric clearance is generally used and is the volume of the space between the piston at dead center and the cylinder head, together with the volume of the steam port at that cylinder end. Clearance thus defined is expressed as a percentage of the piston displacement. Clearance percentages are greater for high-speed, short-stroke engines than for long-stroke, slow-speed engines. *Rotative speeds* greater than 250 r.p.m. are classed *high*, between 150 to 250 r.p.m. *medium*, and less than 150 r.p.m. *slow*.

158. Steam Cycles.—The Carnot and the Rankine cycles of the working fluid are the usual standards by which steam engine performance is judged. Actual engines cannot operate under the Carnot cycle and only approximate the Rankine cycle.

The Carnot Cycle.—Theoretically, the Carnot cycle is the most efficient for engines working between given temperature limits. Fig. 127 illustrates the cycle diagrammatically for the case of an engine using dry and saturated steam. In this cycle the fluid passes through two isothermal and two adiabatic processes. Beginning with a volume of water at A existing at the boiling-point equivalent to the pressure, heat is added to the water along the line of constant pressure and temperature AB, with the result that the water is completely vaporized at the point B. From the point B to C adiabatic expansion occurs with some condensation as the heat content of the steam is reduced because of work done. Reduction of the volume occurs between C and D at constant pressure and temperature with some further condensation of the steam. At D the process changes so that along the line DA adiabatic compression reduces the remaining vapor to water and the cycle starts again at A. The cycle efficiency is dependent upon the absolute temperatures T_1 and T_2 along the isothermal lines AB and CD. The efficiency by the discussion of Art. 19 is

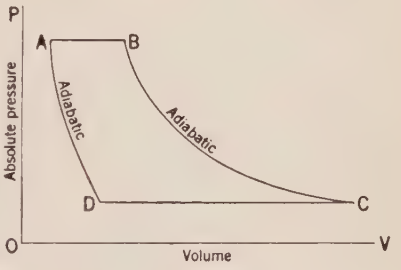


FIG. 127.—Carnot cycle for a steam engine.

$$e_c = \frac{T_1 - T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (117)$$

The cycle is impossible in the commercial steam engine as the water cannot be vaporized and condensed within the engine cylinder.

The Rankine Cycle.—This cycle does not require the vaporization and condensation of water within an engine cylinder. The actual steam engine cycle can approach the Rankine cycle which is taken as the ideal one for steam-engine performance. This ideal standard of performance is built on the premises that the engine operates without clearance, compression of the steam, or pressure losses, and that the steam expansion occurs under adiabatic conditions without heat losses from the cylinder walls.

The Rankine cycle, with complete expansion of the steam, is illustrated diagrammatically by Fig. 128. Steam from an external source is admitted to the cylinder at constant pressure and temperature along the line AB. At B the steam supply to the cylinder is instantly cut off and adiabatic expansion occurs from B to C with a drop in both the pressure and the temperature of the steam. The exhaust valve opens at C so that the steam at constant pressure and temperature is forced out of the cylinder along the line CD. Closure of the exhaust valve occurs at D with the simultaneous opening of the inlet valve and the steam pressure rises instantly along DA.

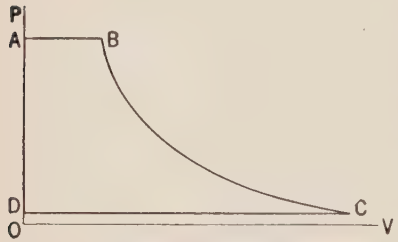


FIG. 128.—Rankine cycle with complete expansion.

Steam engines built to expand the steam completely to a pressure corresponding to the line CD require a larger cylinder volume than is necessary for incomplete expansion of the steam. As the expansion curve BC has a tendency to become tangent to the line CD, the work obtained by steam expansion beyond a certain point is small. The commercial practice is to expand the steam to some pressure E above that of exhaust Fig. 129, and then release it from the cylinder with a rapid pressure drop EF to that of the exhaust line. The small loss of power in the Rankine cycle with incomplete expansion, illustrated by the sectioned portion, Fig. 129, is more than offset by the additional initial

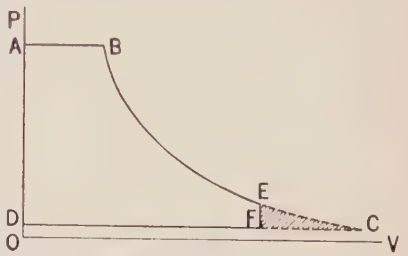


FIG. 129.—Rankine cycle with incomplete expansion.

and operating costs of an engine designed to operate with complete expansion of the steam.

The thermal efficiency of the Rankine cycle is equal to the ratio of the heat available for conversion into work to the total heat supplied.

$$e_R = \frac{H_1 - H_2}{H_1 - h_2} \quad \ldots\ldots\ldots \quad (118)$$

where e_R = Rankine cycle efficiency.
H_1 = total heat of steam initially, B.t.u. per lb.
H_2 = total heat of steam after adiabatic expansion, B.t.u. per lb.
h_2 = heat of the liquid at exhaust pressure, B.t.u. per lb.

159. Theoretical Power.—The engine described in Art. 155 is not the most efficient thermally, but is one of the simplest types of engines built. The discussion of theoretical power developed by such a heat motor is applicable to all classes of engines. Assumptions are made that the conditions of incomplete expansion of the steam prevail, that the operation is without clearance, compression, or reduction of the steam pressure during admission of steam to the cylinder. It is presumed that the valves function instantly and that the equation of the expansion curve is that of a rectangular hyperbola, $PV = C$. Under such conditions the theoretical indicator card or pressure-volume diagram for one end of the cylinder is as illustrated by Fig. 130.

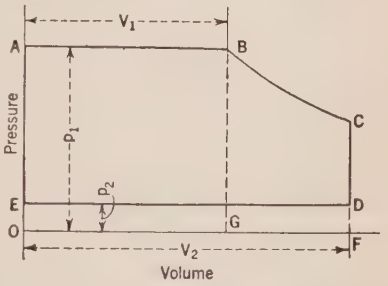


FIG. 130.—Theoretical pressure-volume diagram.

Steam admitted to the cylinder at E enters at a constant pressure p_1 lb. per sq. in. abs. until the point of instant cut-off occurs at B. At this time a volume of v_1 cubic feet of steam has entered the cylinder. Expansion of the steam takes place along the line BC with an increase of volume to v_2. Rapid release at C allows the steam pressure to drop to that of exhaust, p_2 lb. per sq. in. abs. Exhaust occurs at constant pressure p_2 and continues until live steam is again admitted to the cylinder at E.

The total area of the pressure-volume diagram $OABCF$ represents the work done by the steam. The part $OEDF$ depicts the work necessary to force out the steam during exhaust. Consequently, the net work done is represented by the area $EABCD$.

$$EABCD = OABG + BCFG - OEDF$$

$$= p_1v_1 + \int_{v_1}^{v_2} pdv - p_2v_2$$

The work indicated by the net area can be put into an equation of a different form if the expansion ratio $\frac{v_2}{v_1} = r$ is considered, together with the fact that the equation of curve BC is

$$p_1v_1 = p_2v_2 = pv = c$$

hence,

$$p = \frac{p_1v_1}{v}$$

$$EABCD = p_1v_1 + p_1v_1 \int_{v_1}^{v_2} \frac{dv}{v} - p_2v_2$$

$$= p_1v_1\left(1 + \int_{v_1}^{v_2} \frac{dv}{v}\right) - p_2v_2$$

When the integration is made and r is substituted for $\frac{v_2}{v_1}$, the work may be expressed

$$EABCD = p_1v_1(1 + \log_e r) - p_2v_2 \quad . \quad . \quad . \quad . \quad (119)$$

Equation 119 expresses the work done within a single end of the cylinder per stroke of the piston. The mean net steam pressure per stroke pushing the piston forward is the *mean effective pressure* (*m.e.p.*) This pressure is calculated by dividing the work area of the card by the extreme length of the card. Thus, if equation 119 is divided by the card length v_2, the mean effective pressure is

$$p_m = \frac{p_1v_1(1 + \log_e r) - p_2v_2}{v_2} = p_1\left(\frac{1 + \log_e r}{r}\right) - p_2 \quad . \quad (120)$$

The mean effective pressure thus determined is a theoretical value and the actual m.e.p. is always smaller due to certain thermal and mechanical losses which will be discussed later.

The theoretical indicated horsepower developed per end of the cylinder is expressed as

$$i.hp. = \frac{p_mlAN}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (121)$$

p_m = theoretical mean effective pressure, lb. per sq. in.
l = length of stroke, feet.
A = piston area, sq. in.
N = number of strokes per minute.

The total average pressure exerted upon the piston is $p_m \times A$ pounds; and as this force acts through a distance of l feet, the work in foot-pounds done per stroke is $p_m Al$. A working stroke is completed for each cylinder end during a revolution of the crankshaft and flywheel, and the work done per minute is $p_m lAN$. In the case of a double-acting engine, if the cross-section area of the piston rod is neglected, the total power developed by the engine is approximately

$$i.hp. = \frac{2p_m lAN}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (122)$$

Example.—A 7-in. × 10-in. double-acting steam engine is to operate at 200 r.p.m. with the initial steam pressure at 150 lb. per sq. in. abs. and the exhaust pressure at 15 lb. per sq. in. abs. Cut-off is to be at $\frac{5}{8}$ stroke. Clearance and the piston rod are to be neglected. Find the theoretical horsepower of the engine.

Solution.—

$$\text{Cylinder volume} = \left(7^2 \times \frac{\pi}{4} \times 10\right) \div 1728 = 0.222 \text{ cu. ft.}$$

$$\text{Steam volume at cut-off} = 0.222 \times \tfrac{5}{8} = 0.139 \text{ cu. ft.}$$

$$\text{Apparent number of steam expansions} = r = \frac{0.222}{0.139} = 1.60.$$

$$p_m = 150 \frac{(1 + \log_e 1.59)}{1.59} - 15 = 123 \text{ lb. per sq. in.}$$

$$\text{Theoretical i.hp.} = \frac{2 \times 123 \times \frac{10}{12} \times 38.48 \times 200}{33{,}000} = 47.8. \quad \textit{Ans.}$$

160. The Actual Steam Engine Cycle.—Mechanical and thermal limitations in the construction and operation of steam engines do not permit the development of the theoretical power of any engine. Engine clearance volume cannot be eliminated and some compression of the steam entrapped in the clearance is desirable to cushion the reciprocating parts as reversal of the direction of motion occurs at the end of each stroke. This compression of steam lessens the net useful power developed within the cylinder. Losses in steam pressure often occur as the steam flows through the ports to the cylinder and the engine valves do not function instantly at cut-off and release. The reduction of steam pressure caused by friction losses in the ports and the throttling action of valves in opening or closing is known as *wiredrawing*. Also thermal losses occur within the cylinder.

As a result of the foregoing losses, the actual pressure volume diagram obtained from an engine shows less power developed than the

theoretical diagram for the same operating conditions. By the use of an engine indicator a diagram may be taken to show the action of the steam within the cylinder. A sample diagram from a simple slide-valve engine is illustrated by Fig. 131. The beginning and end of the various events of the cycle are marked. Indicator diagrams from an engine more economical in the use of steam are shown in Art. 207. Fig. 168.

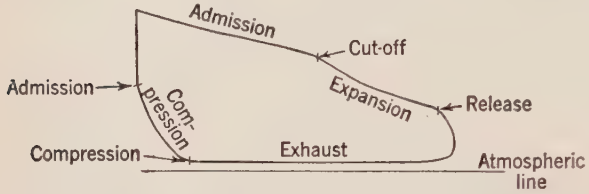


Fig. 131.—Actual pressure-volume diagram.

The relation of the actual to the theoretical diagram is shown graphically, Fig. 132, by properly superimposing the actual diagram over the theoretical or ideal diagram. By means of measurements and computations explained later, the mean effective pressure of the actual diagram can be obtained. The ratio of the mean effective pressure of the actual diagram to the mean effective pressure of the theoretical diagram is the *diagram factor*. This ratio or factor for small simple throttling governor slide-valve engines ranges from 0.58 to 0.77. Engines of better design which expand the steam a greater number of times have diagram factors ranging from 0.80 to 0.90.

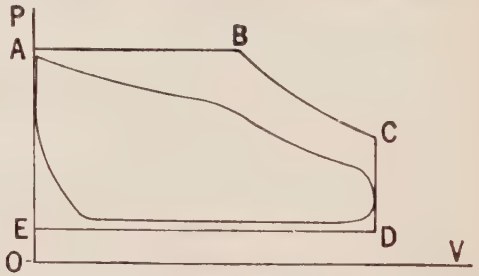


Fig. 132.—Actual and theoretical PV diagrams.

161. Engine Indicators.—Instruments for drawing actual pressure-volume diagrams of the steam cycle are of two general types, inside and outside spring indicators. As the names imply, the springs resisting the movement of the indicator piston are either within the indicator cylinder, Fig. 133, or exterior to it, Fig. 134. The principle of operation of either indicator is the same, although the pencil and piston mechanisms of the two differ somewhat.

Either indicator is attached to suitable piping leading to the clearance spaces at the ends of the engine cylinder. When a cock in the piping connections is open, the indicator piston is subjected to the varying pressures within the engine cylinder. Pressure exerted upon the indicator piston overcomes the resistance of the indicator spring in varying

amounts and causes the piston rod to move. The motion of the piston rod is transmitted through a carefully designed linkage system to give

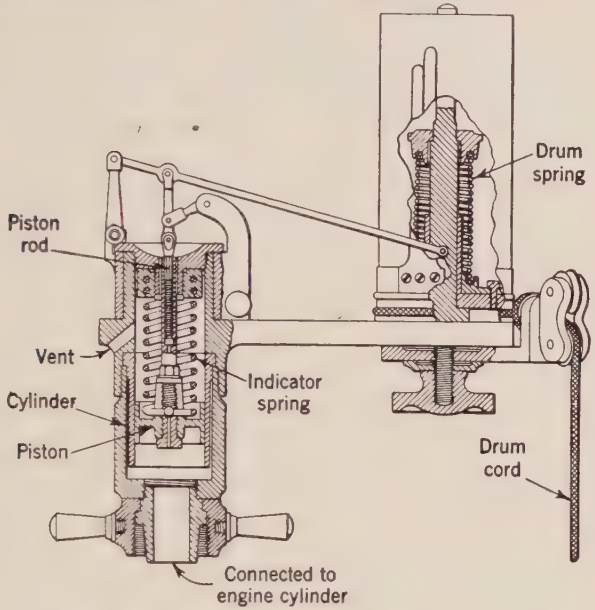


FIG. 133.—Crosby inside-spring indicator.

movement to a lever carrying a pencil point or stylus. The pencil point or stylus moves at all times parallel to the line of motion of the indicator piston. Indicator springs are made and calibrated to allow a vertical movement of the pencil point through a distance of 1 in. when a given pressure in pounds per square inch is applied to the indicator piston. This rating in pounds per square inch for a pencil movement of 1 in. is the indicator *spring scale.*

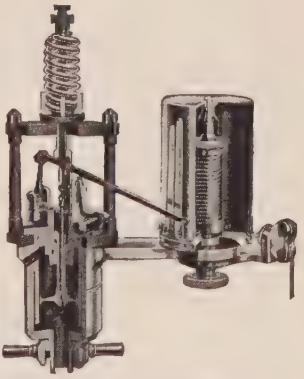


FIG. 134.—Crosby outside-spring indicator.

The pencil or stylus is used to make a record upon a sheet of paper placed about the indicator drum. The drum is rotated forward and backward by a cord wound around the lower part and the action of a coil spring within the drum. The indicator cord should run over a properly located guide pulley and is attached to some part of a reducing motion driven preferably by the engine crosshead.

The function of the reducing motion is to reproduce exactly to a smaller scale the reciprocating motion of the engine piston. Unless the reduction of motion is exact at all points of the piston travel, the pressure-volume diagram produced by the indicator will be inaccurate. Such a condition leads to an erroneous representation of the pressure and volume relations existing within the engine cylinder. A lever type of reducing motion that gives satisfactory results is illustrated by Fig. 135.

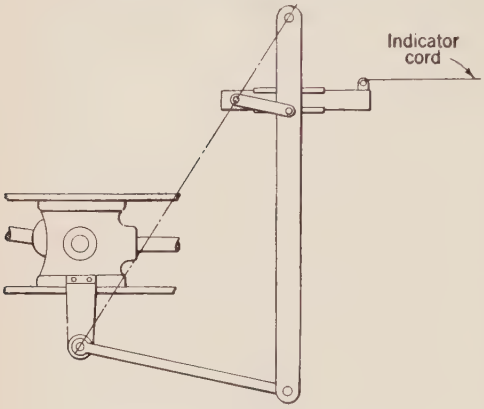


Fig. 135.—Indicator reducing motion.

The pressure-volume diagram obtained by the use of an indicator with an exact reducing motion may be used for the following purposes:

(1) A check of the steam distribution discloses the points at which the steam valve or valves open and close. From this information a study of the adjustments of the valve and the valve gear may be made.
(2) Determination of the average or mean effective pressure existing within the cylinder during a cycle.
(3) Analyses of the actual action of a steam engine.

162. Mean Effective Pressure.—The work areas of indicator diagrams are usually measured by mechanical integrating devices known

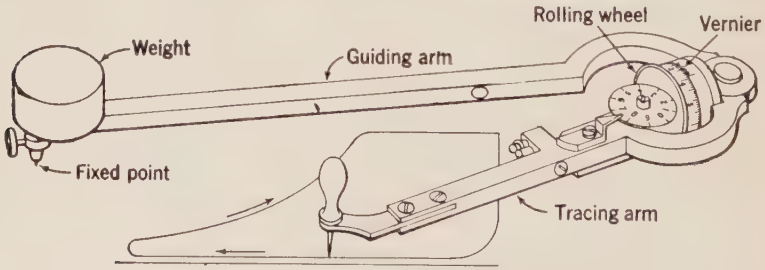


Fig. 136.—Fixed-arm polar planimeter.

as polar planimeters. Such a mechanism, Fig. 136, when properly constructed, adjusted, and handled, will rapidly give area measurements of

sufficient accuracy. When the area of a diagram is known the mean height or mean ordinate O may be determined by dividing the diagram area in square inches by the extreme length in inches of the enclosed area. The mean ordinate in inches is the height of a rectangle whose area is equal to that of the diagram and whose length is equal to the extreme length of the diagram.

The mean effective pressure p_m within the engine cylinder is equal to the product of the mean ordinate O of the indicator diagram, and the scale S of the spring used when the diagram was taken.

$$p_m = O \times S \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (123)$$

163. Indicated Horsepower.—The power produced within the ends of a double-acting steam cylinder is expressed as

$$i.hp., H.E. = \frac{p_m l A N}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (124)$$

$$i.hp., C.E. = \frac{p'_m l A' N}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (125)$$

in which p_m = mean effective pressure, H.E., lb. per sq. in.
p'_m = mean effective pressure, C.E., lb. per sq. in.
l = length of engine stroke, ft.
A = net area of piston, H.E., sq. in.
A' = net area of piston, C.E., sq. in.
N = revolutions per minute.

The quantities $\frac{lA}{33{,}000}$ and $\frac{lA'}{33{,}000}$ of equations 124 and 125 are always fixed for an engine. The ratios when expressed numerically for the cylinder ends are known respectively as the head or crank end *engine constants*. The use of the ratios facilitates the rapid calculation of the power developed when the engine r.p.m. or m.e.p. vary. When a piston rod is attached to the piston in either end of the cylinder, its cross-sectional area must be deducted from the gross piston area to give the net effective piston area.

$$Total\ i.hp. = \frac{p_m l A N}{33{,}000} + \frac{p'_m l A' N}{33{,}000} \quad . \quad . \quad . \quad . \quad (126)$$

When engine cylinders are single-acting, power is not developed in the crank end and the indicated horsepower is that of the head ends alone. The total indicated horsepower of engines having more than one cylinder is the sum of the indicated horsepowers of all of the cylinders.

164. Brake Horsepower.—The actual power output delivered by the crankshaft or at the rim of the flywheel of an engine is the brake horsepower. Due to friction and windage losses occasioned by the moving parts of the engine, the brake horsepower delivered is always less than the indicated horsepower developed.

The brake horsepower of comparatively small engines is determined by the use of a Prony brake, Fig. 137. The brake consists of a flexible band or strap carrying a number of wooden blocks, which encircle the engine flywheel or a brake pulley mounted upon the engine shaft. The wooden blocks are in contact with the surface of the wheel and the strap or band may be tightened or loosened upon the brake wheel by means of a hand adjustment. The brake band is rigidly attached to the two lever arms which meet at a common knife-edge fulcrum C. The horizontal distance from C to the center of the engine shaft is the length r of the brake arm. The free end of the brake arm at C should rest upon a pedestal set upon the platform of weighing scales. This pedestal should be of sufficient height to maintain C at the same elevation as that of the engine shaft center line. The Prony brake is a friction device and the power developed at the rim of the brake wheel is dissipated as heat. Therefore, provision must be made for cooling the rim of the brake pulley. Water is usually used as a cooling medium and is held within the pulley rim by internal rim flanges.

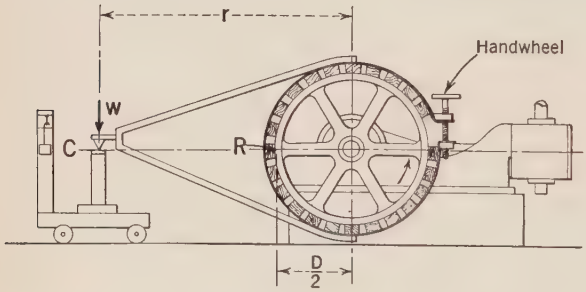


FIG. 137.—Prony brake.

The power absorbed by the brake of Fig. 137 is calculated in the following manner. R is equal to the resistance in pounds offered to motion at the rim of a pulley D feet in diameter. The torque developed at the rim is equal to $R \times \frac{D}{2}$ foot-pounds, and this is also equal to the torque developed at C, or $w \times r$; w being the net pressure applied to the brake pedestal due to the action of the brake and r being the effective brake arm in feet.

$$R \times \frac{D}{2} = wr \quad \text{or} \quad RD = 2wr$$

The force R is applied to the periphery $\pi \times D$ of the wheel which makes N revolutions per minute. The work output per minute is πDRN or $2\pi rwN$ foot-pounds, and

$$b.hp. = \frac{2\pi rwN}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (127)$$

In equation 127 the quantity $\frac{2\pi r}{33{,}000}$ is fixed for a brake and the ratio is the *brake constant.*

Example.—A steam engine equipped with a Prony brake having an arm $5\frac{1}{4}$ ft. long ran at 200 r.p.m. The net load at the end of the arm was 125 lb. Find the brake horsepower.

Solution.—

$$\text{b.hp.} = \frac{2 \times 3.1416 \times 5\frac{1}{4} \times 125 \times 200}{33{,}000} = 25. \quad Ans.$$

165. Friction Horsepower.—The loss of mechanical power within an engine is known as friction horsepower. This quantity is equal to the difference of the total i.hp. and the b.hp. Friction horsepower varies with the percentage of rating developed, size, type, rotative speed, and mechanical construction of an engine. In general, the percentage of friction horsepower is greater in multi-cylinder engines than in simple engines.

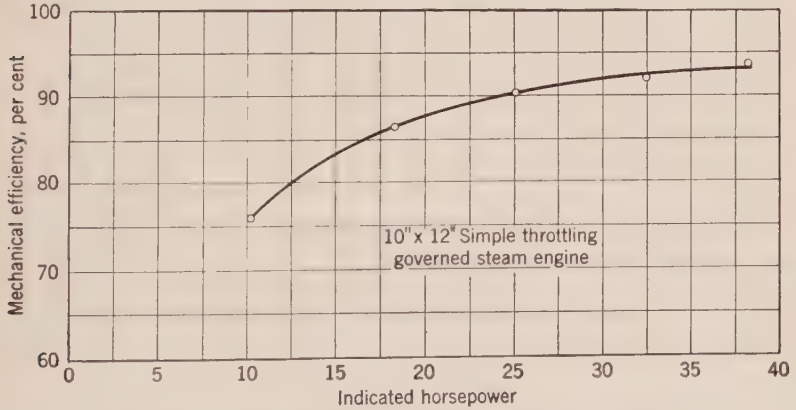


FIG. 138.—Mechanical efficiency of a simple steam engine.

166. Mechanical Efficiency.—The ratio of the engine power output to the power developed within the cylinder (i.hp.) is the mechanical efficiency of an engine.

$$Mechanical\ efficiency = \frac{b.hp.}{i.hp.} \quad . \quad . \quad . \quad . \quad . \quad (128)$$

The mechanical efficiencies of steam engines at normal loads range from 85 to 95 per cent, depending upon the design and workmanship of the engine. Poorly constructed engines may have the lower figure while high-grade machines may give the latter value. The mechanical efficiency of an engine varies with the load upon the engine increasing in amount with the load from light loads to a maximum and then decreasing with a further increase of engine load. Fig. 138 illustrates the variation of mechanical efficiency with engine load in the case of a simple steam engine.

PROBLEMS

1. Find the piston displacement in cubic feet for both the head and crank ends of a 12-in. × 24-in. steam engine having a piston rod $2\frac{1}{4}$ in. in diameter.

2. Find the clearance volume in cubic inches for each end of the cylinder of problem 1 if the clearance percentages are: H. E., 7.89; C. E., 7.04.

3. A 10-in. × 12-in. simple slide-valve engine has a piston rod $1\frac{1}{2}$ in. in diameter. The engine is to operate with a steam pressure of 135 lb. per sq. in. gage at the throttle and exhaust pressure of 0.5 lb. per sq. in. gage in the cylinder. Barometer, 29.6 in. Hg. Find the theoretical mean effective pressure when the cut-off is at 0.6 the stroke.

4. Find the theoretical i.hp. of the engine of problem 3 when the rotative speed is 220 r.p.m.

5. A 10-in. × 10-in. steam engine with a piston rod $1\frac{9}{16}$ in. in diameter gave the following test data: Speed 300 r.p.m., area H.E. card 1.90 sq. in., area of C.E. card 1.95 sq. in., length of cards 3 in., spring scale 80 lb. per sq. in. Find the engine constants and the total i.hp. developed.

6. A 12-in. × 24-in. steam engine with a piston rod $2\frac{1}{4}$ in. in diameter gave the following test data: Speed 100 r.p.m., area H.E. card 3.64 sq. in., area C.E. card 3.76 sq. in., length of cards 4 in., spring scale 80. Find the engine constants and the total i.hp. developed.

7. Find the brake constant for a Prony brake whose arm is 21 in. For a brake whose arm is 5 ft. 3 in. long.

8. The engine of problem 6 was equipped with a Prony brake having an arm 5 ft. 3 in. long. The gross load on the brake scale was 1150 lb. and the brake tare was 250 lb. Find the brake horsepower of the engine.

9. A small steam engine developed 13 i.hp. The engine was fitted with a Prony brake having an arm 21 in. long. The gross brake load was 280 lb. and the brake tare was 80 lb. Find the friction horsepower and the mechanical efficiency of the engine. R.p.m. 170.

10. A steam engine is direct-connected to a 500-kw. electric generator. What is the i.hp. developed within the engine cylinder at full load if the generator efficiency is 94 per cent and that of the engine 92 per cent?

CHAPTER XII

STEAM ENGINE ECONOMY

167. Specific Steam Consumption or Water Rate.—The weight of dry steam consumed by a steam engine per horsepower per hour is the *water rate* of the engine. The expression is made on the basis of either brake or indicated horsepower. The water rate of an engine is dependent upon a number of factors:

(1) Initial steam pressure.
(2) Initial steam temperature.
(3) Exhaust pressure.
(4) Number of expansions of the steam.
(5) Number of cylinders used for expansion.
(6) Method of governing.
(7) Valves and valve gear.
(8) Percentage of rating developed.
(9) Type of engine, counterflow or unaflow.
(10) Mechanical design and construction.
(11) Cylinder insulation.
(12) Steam jackets and reheaters.

Comparative performances of steam engines cannot be judged by the water rates alone because of the number of factors affecting the engine operation. The relative merits of thermal performance of various engines may be obtained more accurately by comparisons of the amounts of heat supplied to the engines per horsepower-hour. Because of the great diversity of operating conditions the water rates of only a few representative engines are given by Fig. 139.

168. Thermal Efficiency.—Thermal efficiency is the ratio of the output (heat equivalent) to the quantity of heat supplied to the engine to produce the power. The ratio or thermal efficiency for saturated steam when computed on the basis of one horsepower is

$$e_a = \frac{2545}{W_s(x_1 L_1 + h_1 - h_2)} \quad . \quad . \quad . \quad . \quad . \quad (129)$$

where e_a = actual thermal efficiency.
2545 = heat equivalent of 1 hp.-hr.
W_s = actual pounds of steam supplied per hp.-hr.
x_1 = quality of the steam supplied.
L_1 = latent heat of vaporization, initial pressure abs., B.t.u. per lb.
h_1 = heat of liquid, initial pressure abs., B.t.u. per lb.
h_2 = heat of liquid, exhaust pressure abs., B.t.u. per lb.

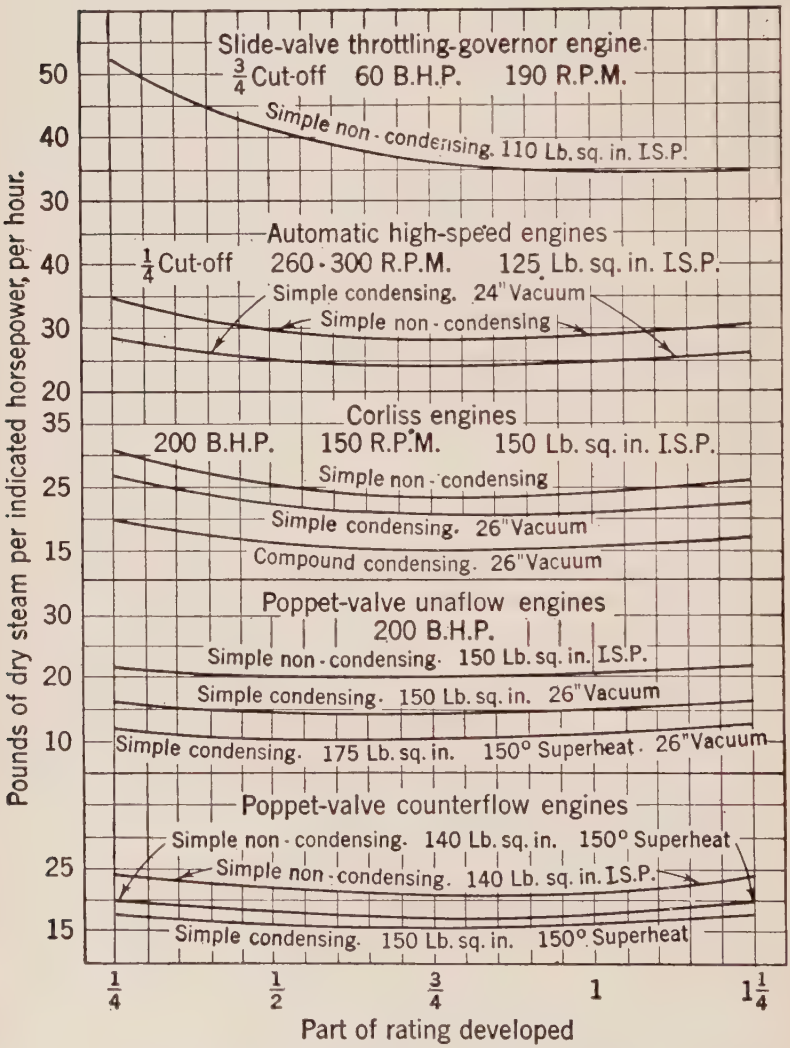


Fig. 139.—Steam consumption of various engines.

When superheated steam is supplied to the engine the expression becomes

$$e_a = \frac{2545}{W_s(H_{sh} - h_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (130)$$

H_{sh} = total heat of superheated steam, initial pressure abs., B.t.u. per lb.

All the heat of the steam supplied to the engine above the heat of the liquid corresponding to the temperature at exhaust pressure is considered as available to the engine.

Inspection of equations 129 and 130 reveals that, in either case, when the initial conditions are unchanged, anything which reduces the quantity W_s increases the thermal efficiency. The thermal efficiency varies inversely as the steam consumption per hp.-hr.

Example.—A steam engine uses 40 lb. of wet steam per b.hp-hr. Initial pressure of the steam is 150 lb. per sq. in. abs., exhaust pressure 15 lb. per sq. in abs., steam quality 98 per cent. Find the thermal efficiency.

Solution.—

$$e_a = \frac{2545}{40[(0.98 \times 864.9) + 329.8 - 181]} = 0.064 \text{ or } 6.4 \text{ per cent.}$$

169. Engine Efficiency.—The ideal or Rankine cycle has been set up as the best possible performance for any conditions. The theoretical efficiency for this cycle was discussed in Art. 158. The actual engine performance falls short of the ideal because the conditions of the ideal cycle cannot be fulfilled. The ratio of the actual thermal efficiency to that of the Rankine cycle for the same pressure and temperature conditions is the *engine efficiency*, e_r.

$$e_r = \frac{e_a}{e_R} = \frac{2545}{W_s(H_1 - h_2)} \times \left(\frac{H_1 - h_2}{H_1 - H_2}\right) = \frac{2545}{W_s(H_1 - H_2)}. \quad (131)$$

H_2 is the total heat per pound of steam after it has been expanded adiabatically from the initial to the final pressure. This heat content may be obtained from the Mollier diagram or it may be calculated as $H_2 = x_2L_2 + h_2$. The final quality of the steam is computed by the method of Art. 38.

The engine efficiency varies from 0.35 for poorly designed and operated engines to 0.70 or 0.80 for high class, high efficiency engines.

170. Thermal Losses.—Certain losses occurring in steam engines affect the steam consumption and the thermal efficiency under operating conditions. These losses include:

(1) Heat of the exhaust steam.
(2) Initial condensation.
(3) Re-evaporation.
(4) Incomplete expansion.
(5) Leakage.
(6) Radiation.
(7) Clearance volume
(8) Wiredrawing.

171. Exhaust Losses.—The largest loss of heat in both the ideal and the actual steam-engine cycle is the quantity of heat rejected by the engine and carried away by the exhaust steam. Cheap and poorly designed engines often discard more than 90 per cent of the total heat supplied. Other engines of better design and construction, together with the aid of auxiliary equipment, are able to reduce the amount of heat thus wasted. The exhaust losses are seldom less than 70 per cent of the total heat of the supplied steam, even with the highest grade of equipment operating under the most favorable conditions.

172. Initial Condensation and Re-evaporation.—Steam entering the cylinder of a counterflow engine strikes comparatively cool port and cylinder walls. The port and cylinder walls are cooled by the expanded exhaust steam, which has its temperature reduced in the process of doing work. The incoming steam at high pressure and temperature immediately gives a part of its heat to the surrounding cool surfaces. The result is that a part of the saturated steam is condensed, or if the steam is superheated the steam temperature is reduced. Condensation of steam represents a heat loss and adds to the difficulty of lubricating the cylinder walls. Next to the heat losses existent in the exhaust, initial condensation is one of the largest losses occurring when saturated steam is used.

A part of the condensation may be re-evaporated later in the stroke as expansion takes place with diminishing pressure of the steam. When re-evaporation occurs the necessary heat is abstracted from the steam of the cylinder. This heat becomes unavailable to the engine and is lost in the exhaust. So long as the interior surfaces of the cylinder remain wet the transfer of heat through them to the outside is augmented.

Initial condensation is affected by the speed of the engine and the temperature range maintained within the cylinder. The slower the engine speed the greater is the amount of time available for the transfer of heat from the steam to the cylinder walls. The more that steam is expanded within a single cylinder, the greater the temperature range and the lower the temperature of the exhaust steam; cooling the walls of

the steam ports and the cylinder. In general, shortening the period of steam admission, by earlier cut-off, improves the steam economy of an engine; the effect produced is more than enough to offset the increased initial condensation losses. However, for single-cylinder engines cut-off earlier than one-fourth of the stroke is not economical as the condensation losses exceed the gains secured by a greater number of expansions of the steam.

173. Incomplete Expansion.—Certain types of valves and valve gears cannot be adjusted to give an early economical cut-off without interference with the other events of the stroke, that is, release, compression, and admission. Under such conditions the number of steam expansions is limited and the economy and thermal efficiency of the engine are reduced.

174. Leakage Losses.—Leakage of steam through the engine valves into the cylinder during the expansion part of the stroke reduces the number of steam expansions within the cylinder. Leakage around the piston into the opposite end, or leakage through the valves into the exhausting end of the cylinder, increases the back pressure against which the piston works. The latter condition represents steam lost to the exhaust without the production of mechanical energy.

175. Radiation Losses.—Unless the cylinder walls and heads are well lagged with insulating materials, a considerable amount of heat will be lost through them by conduction, radiation and convection. Losses due to the same causes also occur from the piston rod and valve rod. One of the effects of these heat losses is to increase the initial condensation of the engine. Although radiation losses cannot be eliminated entirely, they can be reduced materially by using proper precautions.

176. Clearance Volume Losses.—Clearance volume of an engine cannot be eliminated as it is impossible to provide steam and exhaust ports without volume. Furthermore, the piston must not strike the cylinder heads. A graphical representation of clearance is given by Fig. 140 where OG represents the engine clearance and GH the total piston displacement on the pressure-volume diagram. The total volume of the cylinder is equal to the piston displacement GH plus the clearance volume OG. The volume of steam within the cylinder at the point of cut-off B is equal to the sum of the clearance volume

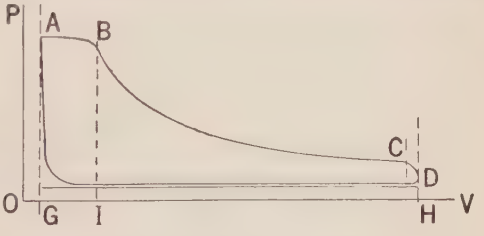


Fig. 140.—Effect of clearance volume.

OG and the volume of piston displacement completed, GI. Therefore, the actual number of expansions occurring in the cylinder is

$$r = \frac{OG + GH}{OG + GI} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (132)$$

The weight of steam required in the cylinder at cut-off is that necessary to fill the clearance at the pressure of point A and that necessary to fill the piston displacement at cut-off. The greater the magnitude of the clearance volume the greater is the weight of steam required to fill this unavoidable space. During exhaust the clearance is filled with low-pressure steam and if the engine has no compression a greater weight of steam is required to fill the clearance at the pressure of point A. Therefore, compression is provided in the actual engine for two reasons:

(1) Reduction of the weight of steam required to fill the clearance volume.
(2) Cushioning of the heavy reciprocating parts.

Steam, when compressed in the clearance space, incurs a rise in temperature due to the work done upon it. When the temperature of the compressed steam exceeds that of the clearance space walls, cooled by the exhaust steam, a heat transfer takes place to the cooler surfaces. A thermal loss similar to that of initial condensation then takes place in the clearance space.

177. Wiredrawing.—Reductions of steam pressure due to the throttling action of valves as they open and close, and the restrictions of steam ports, affect the power output and economy of engines. The principal effect of wiredrawing is to reduce the availability of the heat energy by the pressure loss. Single-valve engines suffer greater losses due to wiredrawing than multi-valve engines.

178. Methods of Increasing Engine Economy.—The following methods are commonly used to improve the economy and increase the thermal efficiency of the steam engine:

(1) Reduced exhaust or back pressure.
(2) Increased initial steam pressure.
(3) Superheating.
(4) Compounding.
(5) Steam jackets.
(6) Improved valve gears.
(7) Reheaters.
(8) Single-flow cylinders.

179. Reduction of Exhaust Pressure.—The pressure of the expanded steam within an engine cylinder during the exhaust stroke is largely controlled by the pressure existing in the exhaust piping attached to the cylinders. Engines may operate with the exhaust pressure above atmospheric or less than atmospheric, depending upon the disposition of the exhaust steam. *Non-condensing* operation allows the steam to be wasted to the atmosphere or to be used at some pressure greater than atmospheric for heating or process work. *Condensing* operation necessitates the reduction of the steam to water within a closed chamber with a consequent reduction of the exhaust steam pressure. Condensing operation is accomplished and facilitated by the use of the equipment described in Chapter XVI.

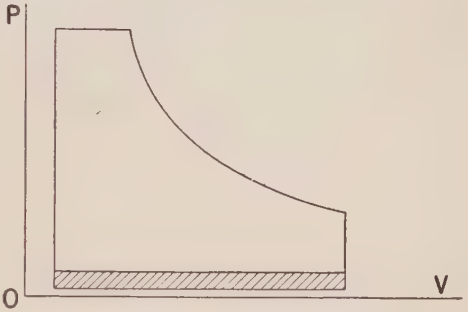


FIG. 141.—Effect of reduced exhaust pressure.

Engines operating with fixed conditions of initial steam pressure, quality and percentage of stroke at cut-off show a gain in the horsepower developed when the exhaust pressure is decreased. The theoretical gain in power, produced within a simple engine cylinder with clearance, by reduction of the back pressure, is shown by the sectioned portion of Fig. 141. The effect is a greater horsepower for a given quantity of cylinder feed steam and a decrease in the water rate of the engine.

If the engine is of a type in which the point of cut-off of the steam can be automatically varied, the amount of the cylinder feed is lessened for a given amount of power developed. The engine will show theoretically a decreased water rate and an increased thermal efficiency. An increase in the exhaust pressure produces the opposite effects. Engines operating with fixed initial steam pressures and facilities for a greater number of expansions of the steam show theoretical diagrams similar to that of Fig. 142, in which the sectioned area represents the gain in useful work or power.

The foregoing theoretical considerations are partially true in the actual engine within certain limits. In single-cylinder engines of the counterflow type, the gain in steam economy is less as the back pressure is reduced. Leakage and initial condensation losses increase especially if the point of cut-off is made earlier in the stroke. Compound engines and simple single flow engines are not affected as much by losses which

tend to mitigate the beneficial effects of the reduced back pressure. In general, for each engine there is a critical back or exhaust pressure at which the engine gives its maximum economy.

Consideration must always be given to the heat consumption of the auxiliary equipment necessary to produce and maintain the reduced back pressure. The expenditure of heat energy in the condensing equipment together with the operating costs may render extreme reductions of the back pressure non-profitable. Unless an engine has specially designed exhaust ports, vacua of more than 22 in. Hg in simple engines and 25–26 in. Hg in multi-cylinder engines are not practical. With the greater vacua largely increased cylinder volumes are necessary. Large cylinders offer mechanical problems as well as greater friction and initial condensation losses.

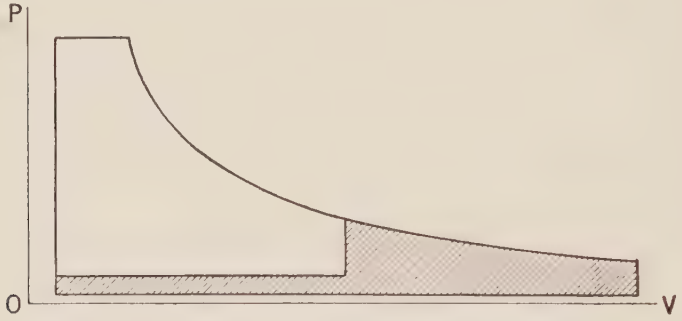


Fig. 142.—Effect of expansion to a lower release pressure.

180. Increased Initial Pressure.—The use of high initial-pressure steam permits more available energy per pound of steam supplied to the engine. The temperature range within the engine may be increased due to the greater initial temperature of high pressure saturated steam. Economy gains due to the use of increased initial steam pressures are not so limited in an engine as the gains secured by the reduction of exhaust pressures.

Modern steam boilers are in operation under a pressure of 1200 lb. per sq. in. Internal-combustion engines of the Diesel type are successfully built to operate with compression pressures of 800 to 1100 lb. per sq. in. Therefore, the design of steam engines to work under such pressures should not be exceptionally difficult. The expansion of extremely high-pressure steam should not be done in a single cylinder if all of the benefits of the high initial pressure are to be secured. The maximum number of steam expansions should be secured and the process should be completed in two or more cylinders. If wide temperature ranges are

maintained in single cylinders the theoretical gains of the increased initial pressure are largely offset by the increased cylinder condensation and other heat losses.

The maximum economies can be effected with pressures varying from 165 to 400 lb. per sq. in. gage. The economy gains with pressures above 400 lb. per sq. in. are of smaller magnitude and often do not warrant the increased initial and operating costs of equipment designed for high pressures. In general, good economy can be obtained with moderate steam pressures and high superheat.

181. Superheating.—In practical applications better results appear attributable to the use of superheated steam than theoretical considerations of the ideal steam engine cycle indicate. The anomaly is due to the fact that initial condensation losses are not considered in the ideal cycle.

The use of superheated steam secures a marked decrease in the cylinder condensation and increased economy of the engine. Engines showing the largest amounts of initial condensation with saturated steam show the greatest savings with superheated steam. Under average conditions the steam consumption of an engine using saturated steam is reduced 1 per cent for each 10 deg. of superheat when superheated steam is used. Single-cylinder counterflow engines show the greatest gain in economy by the use of superheated steam.

Highly superheated steam may offer some disadvantages in the lubrication as well as warping and distortion of the rotating or sliding valves of certain types of engines. These troubles can be eliminated by the use of poppet valves for the admission of steam to the cylinder.

182. Compounding.—Cylinder condensation has been shown to be one of the important thermal losses of the steam engine. This loss is especially prone to occur in single-cylinder engines having early cut-offs and a large number of steam expansions in the one cylinder. Initial condensation losses can be reduced by dividing the number of expansions of the steam between two or more cylinders. In such an arrangement the pressure and temperature ranges, as well as clearance losses within the cylinders, are reduced and better steam economy is secured in the counterflow type of engine.

When the pressure and temperature ranges are divided between two cylinders the engine is designated as *compound*. Engines expanding steam successively in three cylinders are known as *triple-expansion* and those using four successive cylinders for the steam expansion are called *quadruple-expansion* engines. Engines which use multi-cylinders for expanding steam through a large pressure and temperature range are able to employ high initial steam pressures and the minimum feasible

back pressures. Non-condensing compound engines are not desirable. The minimum initial steam pressures for engines having more than one cylinder for the progressive expansion of the steam are:

Compound	125 lb. per sq. in.
Triple-expansion	200 lb. per sq. in.
Quadruple-expansion	250 lb. per sq. in.

The advantages of compounding are: (1) greater range of steam expansion, (2) decreased cylinder condensation, (3) reduction of leakage and clearance losses, (4) possibility of more uniform power application to the crankshaft, and (5) lessened steam consumption per horsepower-hour. The foregoing beneficial effects are partially nullified by: (1) the increased initial cost, (2) multiplicity of parts, (3) increased floor space required, (4) decreased mechanical efficiency and (5) increased heat losses by radiation.

183. Steam Jackets.—In some cases the cylinder walls and heads are made double and the space thus enclosed within them is filled with live steam at boiler pressure. Such arrangements are termed *steam jackets*. The function of steam jackets is to reduce the heat losses from the cylinder walls and heads and thus curtail cylinder condensation. In order to be effective the steam jackets must be free from air pockets, free from condensed steam, and must have unrestricted steam circulation. Steam-jacketed engines have shown improved economy in some cases and in others no improvement. Jackets are used to reduce the losses that superheat also tends to eliminate. The use of superheat is the better method of economy improvement and jackets are not necessary or desirable where superheated steam is employed.

184. Improved Valve Gears.—Simple engines operating with valves and valve gears which do not permit early cut-off are not as economical as those in which the steam may be expanded a greater number of times. Furthermore, in such engines there is often a considerable amount of wiredrawing due to the restrictions of the steam passages and the sluggish action of the valve in opening and closing the ports. Valve gears designed to give early cut-off and quick action of the steam valves tend to improve the thermal efficiency of the unit. Also, by the use of more than one valve the length of the port passages may be shortened and the exhaust steam discharged from the cylinder through separate exhaust ports. The latter arrangement helps somewhat to reduce initial condensation losses.

185. Reheaters.—Multi-cylinder engines in which the steam is successively expanded in two or more cylinders usually have steam storage spaces or receivers between the cylinders. These receivers may

be fitted with pipe coils through which high-pressure steam is circulated. The function of the heating coil within a receiver is to evaporate the moisture of the exhaust steam from the preceding cylinder and if possible to impart to the steam some superheat. Reheating is attempted in an effort to reduce the initial condensation losses in the cylinder following the receiver. If reheating is properly accomplished the effects in the low pressure cylinders of multiple-expansion engines are the same as those of superheated steam in single-cylinder engines. Ample reheating eliminates the necessity for low-pressure cylinder steam jackets.

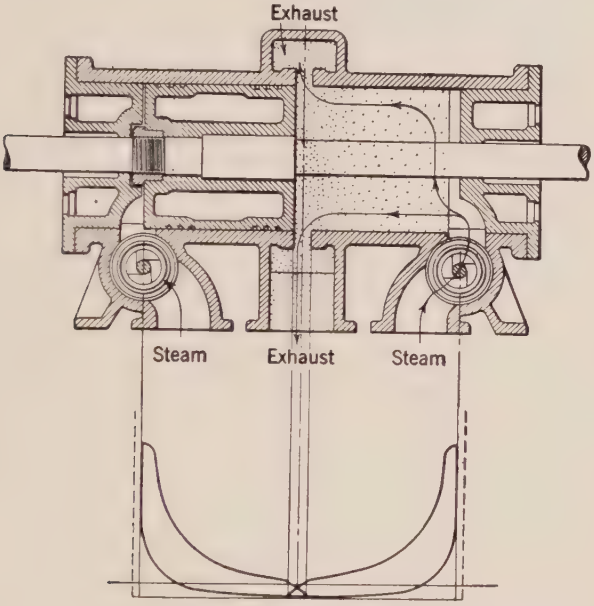


FIG. 143.—Diagrammatic section of a unaflow engine cylinder.

186. Single-Flow Cylinders.—The counterflow type of engine operates at a disadvantage due to the fact that all of the expanded and cooled steam is returned to the inlet end of the cylinder during exhaust. This cooled steam chills the cylinder walls and in many cases the port passages through which the hot live steam enters the cylinder. The result is that large initial condensation losses are produced. The single direction or unaflow engine reduces this loss to a marked extent. The reduction of condensation is accomplished by having separate inlet valves for the live steam and short port passages at each end of the cylinder, Fig. 143. The exhaust ports of the engine are located in the walls of the cylinder barrel midway between the cylinder ends. The engine

piston serves as the exhaust valve and alternately covers and uncovers the ports for exhaust from either end of the cylinder. The path of the steam in passing through the cylinder is in a single general direction and thus cylinder condensation losses are reduced. Single-cylinder unaflow engines are able to show better steam economy than compound counterflow multi-cylinder engines. One disadvantage incurred by these engines is the high-compression pressure developed, especially when operated non-condensing. This trouble is usually largely mitigated by the use of special auxiliary exhaust valves which are held open until near the end of the return stroke. This type of engine is described in more detail in Art. 211, Chapter XIV.

PROBLEMS

1. A steam engine developed 100 b.hp. during a test which gave the following data: Initial steam pressure 135 lb. per sq. in., barometer 29.6 in. Hg, steam quality 99 per cent, total wet steam used per hour 3450 lb., exhaust pressure atmospheric. Find the water rate and the thermal efficiency based on b.hp.

2. If the engine of problem 1 operated with a mechanical efficiency of 92 per cent, what were the water rate and the thermal efficiency based on i.hp.?

3. A simple engine operates with the exhaust into a vacuum of 20 in. of mercury, barometer 29 in. Hg, initial steam pressure 140 lb. per sq. in. abs., steam with 150 deg. fahr. superheat. When operated with a load of 300 b.hp. the total steam consumption per hour was 6250 lb. Find the water rate and the thermal efficiency based on b.hp.

4. Find the Rankine cycle efficiency in problem 1.

5. Find the Rankine cycle efficiency for the data of problem 3.

6. Find the engine efficiency for the data of problem 1.

7. Find the engine efficiency for the data of problem 3.

CHAPTER XIII

VALVES, VALVE GEARS, AND GOVERNORS

SLIDE-VALVE ENGINES

187. Functions of Valves.—The purpose of steam engine valves and the gear which actuates them is to control the distribution of steam within the engine cylinder. The distribution process includes the control of the events of admission, cut-off, expansion, release, exhaust, and compression of the steam during each working cycle. Both the power developed by the engine, and the economy of operation are affected by the valve and the valve-operating mechanism. Certain valves, together with their gear, have inherent limitations which materially reduce the thermal efficiency of the engine thus equipped. Other engines equipped with a better valve mechanism and operating under identical conditions, show considerable savings in the amount of heat energy that must be supplied to produce a given amount of mechanical power.

188. The D Slide-Valve.—One of the simplest and easiest constructed of all steam engine valves is the D slide-valve. The designa-

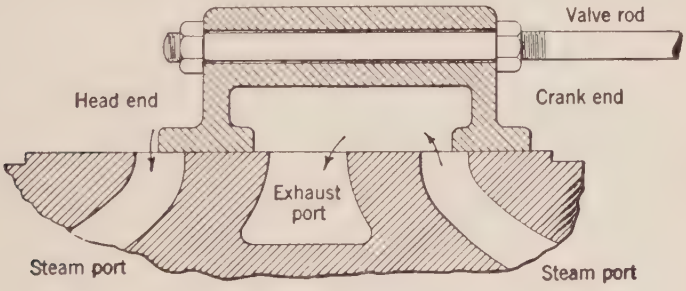


FIG. 144.—Simple D slide-valve.

tion D is applied because of the sectional shape of the valve as shown by Figs. 126 and 144. The valves as illustrated are allowing admission of steam to the head end and exhaust of steam from the crank end of the cylinder.

Such valves are used in engines of simple design and where the first cost of the engine is of prime importance. Valves of this type, when used

singly, do not permit cut-off within the engine cylinder at less than about five-eighths of the stroke without serious interference with the other events of the cycle. If attempts are made to secure earlier cut-off with a single valve controlling all of the events, the power output is reduced and smooth operation of the engine is prevented. The comparatively late cut-off thus secured limits the number of times that steam may be expanded within a single cylinder and reduced heat economy of the engine results.

Where exhaust steam can be used for feedwater heating, process work, or the heating of buildings, engines equipped with slide valves and operating with high water rates may be used with economical results. The engine producing mechanical power acts as a rotary pressure-reducing valve and the heat rejected to the exhaust is utilized for some useful purpose. When disposal of exhaust steam is made in this manner, mechanical power may be secured at reduced costs because of the market value of the exhaust steam. The assumption must not be made that the mechanical power secured is produced without cost even though live steam from the boilers would be required for the heating demands if exhaust steam were not available. The power production must be charged with certain initial, maintenance, interest, depreciation and operating expenses.

Definite terms of nomenclature are applied to all valves. These terms are most readily explained by the use of sections of a D valve, but the terms of the following articles are applicable to other valves.

189. Lead.—Admission of steam to the cylinder usually occurs just prior to the dead-center position of the piston. With admission ahead of dead-center position the valve uncovers the port near the end of the piston stroke. Lead is the amount by which the valve uncovers the steam port when the engine is at the dead-center position, Fig. 145. The measurement is made in terms of inches. Lead is given to an engine to aid in cushioning the piston as it comes to rest, and to reduce wiredrawing losses as the piston starts on its outward stroke.

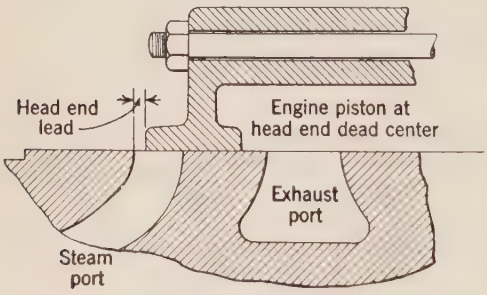


Fig. 145.—Valve lead.

Leads may be made equal at each end of the cylinder by adjustment of the valve gear. When the leads are equal at both ends of the cylinder the cut-offs are unequal, with the ordinary slide-valve gear.

190. Lap.—Two designations of lap are made, which are *steam lap* and *exhaust lap*. Measurements of steam or exhaust laps cannot be correctly made until the valve is in its mid-position of travel. A valve is in mid-position when it is half way between the extremities of its travel. Mid-position of the valve changes with every change of the valve setting and the valve is not always in mid-position when its center line coincides with the center line of the valve seat.

Steam lap is the amount by which the admission edge of the valve overlaps the admission edge of the port when the valve is in mid-position, Fig. 146. Exhaust lap is the amount by which the release edge of the

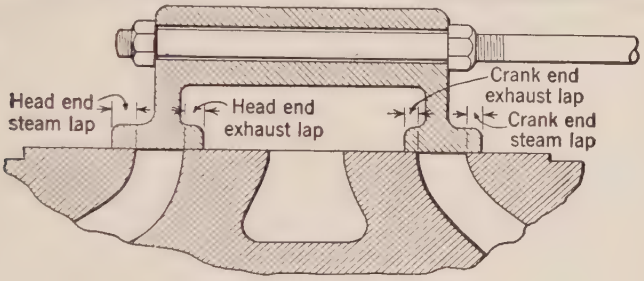


Fig. 146.—Steam and exhaust laps.

valve overlaps the release edge of the port when the valve is in mid-position, Fig. 146. The steam and the exhaust laps are not usually equal for the two-cylinder ends, especially if the cut-offs are equal. The laps are necessarily unequal due to the angularity of the connecting rod and the eccentric rod. Greater travel of the piston and the valve occur in the first quarter of the revolution, in which the piston moves from the head end. The angularity of the eccentric rod is of small effect, but that of the connecting rod is appreciable. This angularity of the connecting rod requires that a greater steam lap be given to the head end of the valve than is given to the crank end.

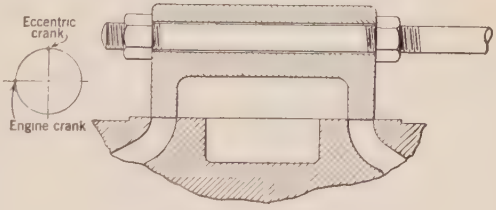


Fig. 147.—Slide-valve without laps.

Steam laps are necessary in order to give cut-offs earlier than the end of the stroke. A valve without lead or laps is shown by Fig. 147, and cut-off occurs at the end of the stroke. Valves of this type are frequently used in direct-acting pumps, Fig. 219.

Exhaust laps aid in giving compression to the steam near the end of the return stroke, although some automatic high-speed engine valves

have no exhaust lap. In some cases the valve does not completely cover the port on the exhaust side when in mid-position. This latter condition is called *exhaust clearance.*

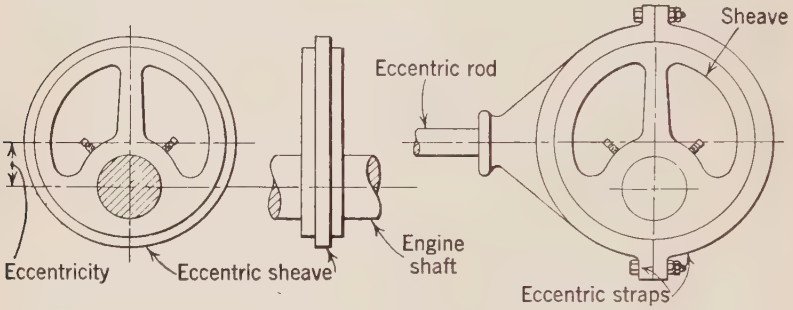


Fig. 148.—Eccentric sheave and straps.

191. Angle of Advance.—The effect of an eccentric, Fig. 148, is that of a crank which moves the valve back and forth as the eccentric is rotated. This effect is obtained by making the eccentric in the form of an enlarged crank pin whose center does not coincide with that of the shaft upon which it is mounted. The eccentric turns about the center of the crankshaft and not about its own geometrical center. The distance between the center of the crankshaft and the center of the eccentric is called the *eccentricity,* Fig. 148, and is equal to the crank arm effect of the eccentric. The motion imparted to the valve slide and valve is equal to twice the eccentricity.

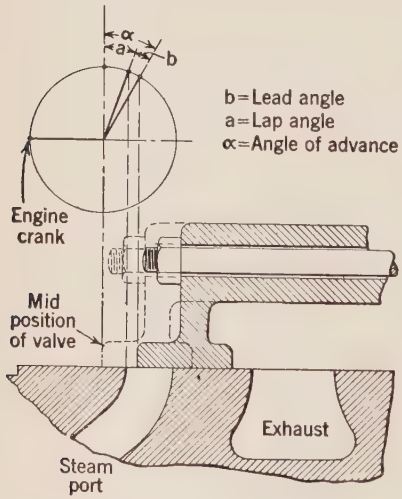


Fig. 149.—Angle of advance.

If the valve had no laps and no lead, the effective crank arm of the eccentric would be set ahead of the engine crank by an angle of 90 deg. However, the valve must be moved to give the necessary lead at dead center which necessitates that the eccentric be ahead of the crank by an angle equal to (90 deg. + α), Fig. 149. The angle a is the lap angle, as the eccentric must be advanced by this amount to overcome the steam lap. As the valve also has steam lead, in order to give the proper timing of events in the cycle,

the eccentric is turned ahead of the crank by a further angle of b, which is the lead angle. The angle α, which is the sum of two angles a and b, is the angle of advance. The angle of advance may also be stated as the total angle between the eccentric crank and the engine crank minus 90 deg.

192. Slide-Valve Setting.—Either of two methods may be employed in adjusting a valve gear to give the best distribution of steam within an engine cylinder. The adjustments may be made in conjunction with the use of an engine indicator or they may be made by measurement. Valve setting by use of an indicator, is simpler as all adjustments are made outside of the steam chest which is closed. The adjustments consist merely in lengthening or shortening the valve rod and changing the eccentric position. Indicator diagrams disclose the effect of the changes made, and when the adjustments are made to give the best diagram possible, the setting is completed.

193. Effects of Slide-Valve Gear Changes.—When the engine has a simple valve gear as shown by Fig. 125, the eccentric crank is set ahead of the engine crank, in the running direction, by an angle of (90 deg. $+ \alpha$), where α is the angle of advance. The effect of advancing the eccentric or making α greater is to make all of the events of the cycle come earlier for both ends of the cylinder. Retarding the eccentric has the opposite effect or makes all events later in the cycle. The adjustments made upon the valve consist merely in lengthening or shortening the valve rod. This change may be made by moving the valve on the valve rod or by changing the effective length of the valve rod at the valve-rod slide and guide. Any adjustment of the valve or valve rod which moves the valve toward the crank end of the cylinder shortens the valve rod. Any change of the valve position toward the head end of the cylinder has the effect of lengthening the valve rod. Irrespective of how the changes of valve position are accomplished the alterations thus produced will hereafter be designated as having been accomplished by lengthening or shortening the valve rod. An inspection of the items of Table XII reveals that changing the length of the valve rod produces opposite effects at the two ends of the cylinder.

194. Location of Engine Dead Centers.—Engine dead-center positions can be rapidly located by the use of a trammel rod, Fig. 150. The engine piston is placed slightly ahead of dead center and corresponding marks are made on the crosshead and one of the guides. Using some convenient point on the engine frame or floor as a reference point, a mark is scribed by the trammel on the engine flywheel. The engine piston is then moved in its running direction beyond the dead center

position until the marks on the crosshead and guide coincide. Another arc is now scribed on the flywheel by the trammel from the same reference point. The linear distance on the flywheel between the scribed arcs is bisected by the aid of dividers and the mid-position thus secured is brought under the free end of the trammel whose opposite end rests on the original reference mark. In locating dead centers the engine should always be turned in its running direction to avoid any lost motion in the moving parts.

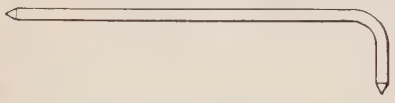


FIG. 150.—Trammel rod.

TABLE XII

EFFECTS OF CHANGES IN LENGTH OF VALVE ROD

Change	Events Made		Lead		Laps	
	Later	Earlier	Greater	Less	Greater	Less
Valve Rod Lengthened	H. E., Adm. H. E., Comp. C. E., C. O. C. E., Release	H. E., C. O. H. E., Release C. E., Adm. C. E., Comp.	C. E.	H. E.	H. E. Steam C. E. Exhaust	C. E. Steam H. E. Exhaust
Valve Rod Shortened	C. E., Adm. C. E., Comp. H. E., C. O. H. E., Release	C. E., C. O. C. E., Release H. E., Adm. H. E., Comp.	H. E.	C. E.	C. E. Steam H. E. Exhaust	H. E. Steam C. E. Exhaust

195. Slide-Valve Setting by Measurement.—Valve setting by measurement is often employed when an engine indicator together with a reducing motion are not available. The first step in the procedure is to locate accurately both the head-end and crank-end dead centers. The linear distance between the two dead-center positions is the actual stroke of the engine. The valve setting by measurement may be made for equal cut-offs or equal leads, but not both at the same time.

Procedure for equal cut-offs.—Lay off from both the head and the crank ends on the crosshead guide linear distances representing the percentages of the stroke desired at cut-off. These measurements should be laid off from the dead center in the direction of the piston motion. Put the engine on dead center and shift the eccentric so that the angle of advance is about 45 deg. Turn the engine in a running direction until the desired portion of the stroke at cut-off is completed. Remove the steam chest cover and lengthen or shorten the valve rod until the cutting-off edge of the valve just coincides with the cutting-off edge of the port at the cyl-

inder end under consideration. Turn the engine in the running direction until the piston is at the desired cut-off position at the opposite end of the cylinder. Measure the amount by which the valve covers or fails to cover the cut-off edge of the port for this end of the cylinder. Lengthen or shorten the valve rod, as the case may be, to correct one-half of the error, and shift the eccentric on the shaft to eliminate the other part of the error. Put the engine piston at the cut-off position at the opposite end of the cylinder, and remove one-half of any error by adjusting the length of the valve rod and the other half by moving the eccentric on the shaft.

Set the engine first on one dead center and then the other, and measure the lead in each case. Measure the position of the valve center on the seat when the valve is in an extreme position of movement. From prior measurements of the valve, the valve seat, and the port widths calculate the head- and crank-end steam and exhaust laps as well as the maximum port openings for each end of the cylinder. See that all nuts are locked tightly and replace the steam chest cover. If an indicator is available apply a normal load to the engine, and take a diagram for each end of the cylinder. Make such minor adjustments as are necessary.

Setting for equal leads.—Place the engine on dead center with the angle of advance approximately 45 deg. Move the valve to give the desired lead. Place the engine on the opposite dead center and secure the desired lead by lengthening or shortening the valve rod to remove one-half the error and shifting the eccentric to correct the other half of the error. Check the lead again at the opposite end of the cylinder and remove one-half of any error by moving the valve and the eccentric, respectively. Find the percentages of the stroke at which the cut-offs occur as well as the steam and exhaust laps and maximum port openings.

196. Balanced Valves.—The slide valve shown by Fig. 144 is unbalanced in that the pressure within the steam chest forces the valve against the seat upon which it moves. Under such conditions a considerable amount of power may be required to move the valve (especially large valves) and difficulty is encountered in the lubrication of the faces of the valve and the valve seat.

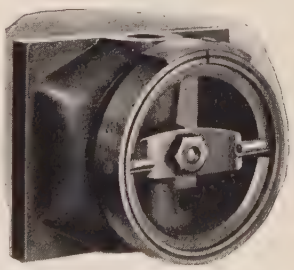


FIG. 151.—Partially balanced slide-valve.

The steam pressure on the back of the valve may be partially relieved in several ways. One method of pressure relief is as shown by Fig. 151 where the valve is made to slide between the steam chest cover and the seat. A

packing ring is fitted to the back of the valve and held against the steam chest cover by the action of a spring. The steam pressure is thus relieved from the greater portion of the back of the valve and the frictional load on the valve gear reduced. The valve thus fitted is only partially balanced, as sufficient pressure is maintained on the back of the valve to hold it firmly against the valve seat.

197. Piston Valves.—The piston type of valve, Fig. 152, is in principle a D slide-valve. Valves of the piston type are balanced as the steam pressure is equally exerted upon all sides. Piston valves move back and forth in a cylindrical valve chamber which confines them. This confinement prevents their lifting from the seat when compression

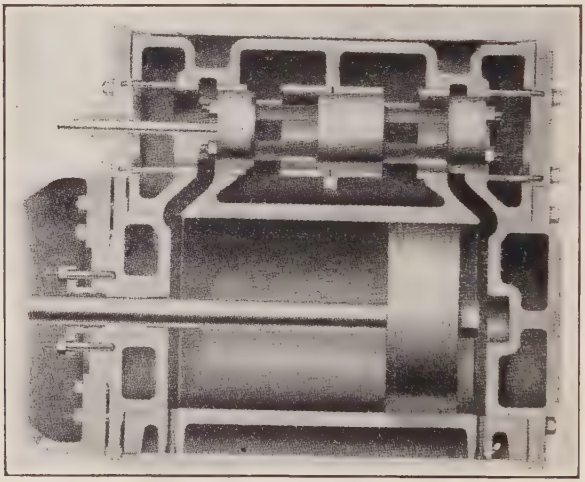


FIG. 152.—Piston valve.

of an excessive amount of water in the clearance space of the engine occurs. Damage to the engine may result under such conditions.

198. Double Ported Valves.—Single steam ports at the cylinder ends may require excessive valve travel in order that the port be fully opened during the admission period. Sluggish valve action may cause considerable wiredrawing during the admission period. In order to eliminate such action, double steam ports to the cylinder ends or ports through the valve or both may be provided. A ported valve is shown by Fig. 153, which illustrates a Sweet valve. The valve illustrated allows by a small movement a greater port opening. The Sweet valve is also partially balanced as it moves between a pressure plate and the valve seat.

199. Speed Regulation.—Ordinarily steam engines are provided with a governing mechanism which functions to maintain nearly a con-

stant speed of the engine. The purpose of the governor is to limit or increase the power input of the engine as the load decreases or increases. The regulation must be very close in certain classes of work, as for instance, engine-driven electrical generators where the voltage varies directly as the engine speed. Governors must also serve to limit excessive and dangerous engine speeds.

A governor must be stable in order to give satisfactory regulation. The term stable signifies that for any predetermined speed the governor will assume a definite position for that speed. Any variation from the predetermined speed will cause the governor to move to bring the speed back to normal. The governor movements must be regular and certain without excessive speed fluctuations. Unstable governors move rapidly from one speed to another without being able to control the engine at any fixed rotative speed. When the governor is over-sensitive and the engine speed varies rapidly from low to high and then from high to low, the engine is said to "hunt."

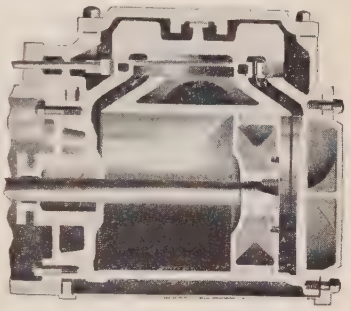


FIG. 153.—Sweet valve. Ames steam engine.

Engines are built to deliver a normal or full load at some designated rotative speed, and the governor should be adjusted to give the normal speed. The *coefficient of regulation* is the percentage of variation of the rotative speed from the normal or full load speed. The coefficient of regulation expressed as a percentage is

$$K_R = \frac{(N' - N)100}{N} \quad . \quad . \quad . \quad . \quad . \quad . \quad (133)$$

in which K_R = coefficient of speed regulation, per cent.
N = normal full load speed, r.p.m.
N' = no load speed, r.p.m.

For electric lighting service K_R should not exceed $1\frac{1}{2}$ per cent for slow changes, and 2 per cent for sudden load changes.

Speed regulation of engines is accomplished by two methods of governing: *throttling* and *cut-off*. The former method effects the regulation by varying the initial steam pressure, while the latter is often but not always accomplished by some form of a shaft or flywheel governor, which varies the volume of steam admitted to the cylinder. Releasing gear Corliss engines have variable cut-off without shaft governors.

200. Throttling Governor.—This type of governor is employed with reciprocating engines having a fixed point of cut-off, small steam turbines, and air compressors.

The construction of a typical throttling governor is shown by Fig. 154. A balanced valve A is carried upon the lower end of the valve spindle B, which passes upward through a stuffing box at the top of the governor body. The upper end of the spindle B is rigidly attached to a speed-changing device C. Above the speed-changing device is another spindle,

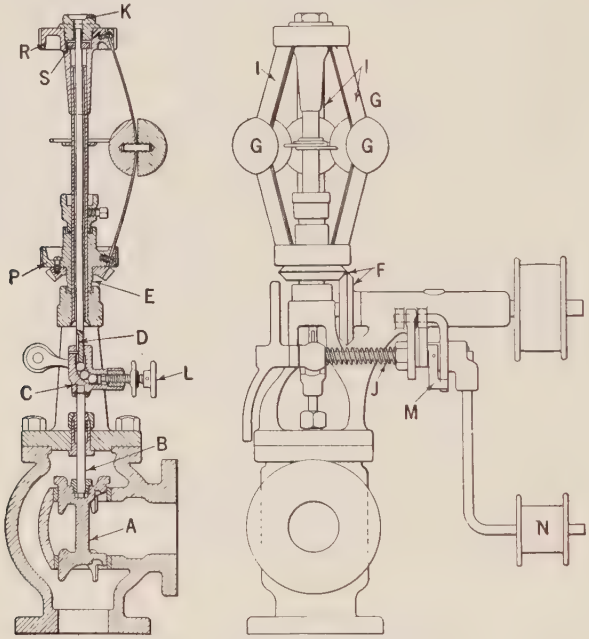


Fig. 154.—Pickering throttling governor.

D, which passes upward through a revolving sleeve E held within a vertical bearing. The sleeve is driven through a pair of miter gears F, F, from a short shaft supported by a bracket. The rotative motion of the shaft is derived from the main shaft of the engine by means of a belt and pulleys. The upper stem D is centered in its contact with the speed changer and does not revolve.

Governor balls or weights, G, G, G, each made in two pieces, are fastened to three flat steel springs, I, I, I, by means of a central stud. The springs are rigidly attached to the revolving sleeves P and R, making a unit that actuates the valve. The cap K, screwed into the recess of the revolving sleeve R, bears on the upper valve stem button S and is em-

ployed for depressing the governor valve. When the engine is not running, the balanced valve A is held open by the action of a coil spring J, which may be adjusted for various spring tensions, according to the individual conditions of the engine.

The action of the governor is procured as the revolving flyballs are thrown outward by centrifugal forces. The movement thus secured causes the cap K to push the stem D downward, and thus partially close the valve A against the action of the coil spring J. This action causes a restriction in the amount and the pressure of the steam admitted to the steam chest of the engine or turbine. A further increase of engine speed causes the governor balls to rotate more rapidly and to be moved farther outward, thus further closing the valve A against the action of the spring. A decrease of engine speed has the opposite effect.

Engine speed changes may be effected by means of an adjusting screw L, which allows hardened steel balls to be inserted under or removed from directly underneath spindle D. The effect of the balls is to lengthen or shorten the stem D and thus cause the valve A to be opened or closed more or less for a given position of the governor weights or flyballs. The governor is fitted with a safety device M, so that if the governor belt should break the idler pulley or belt tightener N will drop and by means of a ratchet and pawl arrangement will close the valve A and thus stop the engine or turbine. The effect of a throttling governor with varying engine loads is shown by Fig. 155.

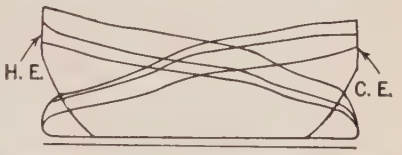


FIG. 155.—Action of a throttling governor.

201. Flywheel or Shaft Governors.—Engines which have their speed controlled by the duration of the steam admission period usually have cut-off for full or normal loads at $\frac{1}{4}$ to $\frac{1}{3}$ the stroke. When speed increases occur due to decreased load, the governor functions to make the cut-off earlier and when heavy loads decrease the speed cut-off comes much later in the stroke, Fig. 156. The control of the point of cut-off is often accomplished by shaft or flywheel governors of the following types:

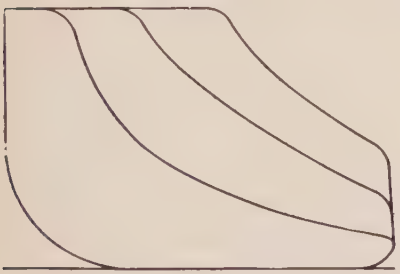


FIG. 156.—Action of a cut-off governor.

(1) Centrifugal.
(2) Inertia.
(3) Combination of centrifugal and inertia.

As a matter of fact, it is very difficult to build a flywheel governor that does not embody some of the characteristic forces of both the centrifugal and inertia types.

Flywheel governors vary the point of cut-off by changing:

(1) The angle of advance of the eccentric.
(2) The eccentricity of the eccentric and the amount of valve travel.
(3) Both the angle of advance and the valve travel.

202. Centrifugal Governor.—Fig. 157 illustrates a Robb-Armstrong-Sweet centrifugal governor as built into the flywheel of an Ames single valve, center-crank steam engine. The mechanism consists of a leaf spring rigidly attached at one end to a spoke of the flywheel. The free end of the spring carries a weight which is connected by a link to a lever weighted at one end. The weighted lever is pivoted near the flywheel hub at about the middle of the lever. Between the lever weight and the lever pivot point an eccentric rod is attached to the lever. This eccentric rod transmits motion to a valve rocker and thus gives motion to the engine valve.

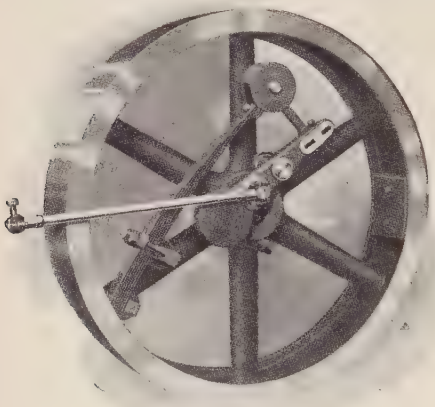


FIG. 157.—Robb-Armstrong-Sweet centrifugal-inertia governor, Ames steam engine.

In the position shown, the governor weights rest near the center of the wheel, due to the action of the spring. As the engine acquires speed, centrifugal forces move the weights outward against the spring resistance. When the weight movement occurs the weighted lever is moved about its pivot and the point of attachment of the eccentric rod is shifted with respect to the center of the engine crankshaft. The action of the governor is to change both the angle of advance and the amount of the valve travel. The operating speed of the engine is changed by moving the point of attachment of the link at the weighted lever.

203. Inertia Governor.—A simple form of inertia governor is shown by Fig. 158, which illustrates the flywheel and governor of an Elliott engine. The governor embodies a weighted lever pivoted at its center

on one of the flywheel spokes. When the engine is at rest, the action of a coil spring brings one of the weights against a stop as shown at the top of the wheel. A swinging eccentric is rigidly attached to the weight lever and rotates with the lever about the pivot point. The engine shaft passes through a slot of the eccentric; so that the eccentric is entirely free from the engine shaft.

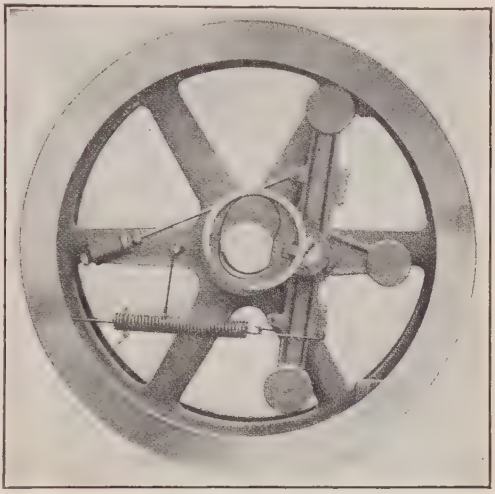


FIG. 158.—Inertia governor of an Elliott steam engine.

When the engine is at rest the eccentric is at a position necessary to give the maximum cut-off. As the engine speeds up, the rim of the flywheel tends to run ahead of the governor weights which lag behind due to inertia forces. The result is that the eccentric position is shifted with respect to the engine crank and the point of cut-off is made earlier.

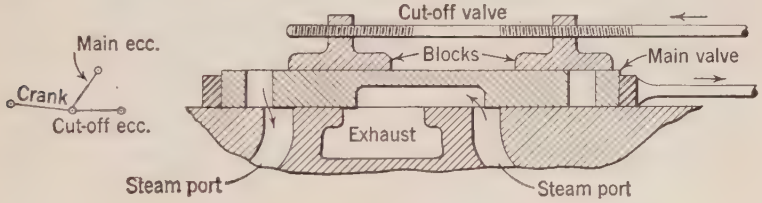


FIG. 159.—Riding cut-off valve at admission.

When the flywheel tends to run slower the governor weights crowd ahead, due to inertia forces, and the eccentric is shifted to give later cut-off.

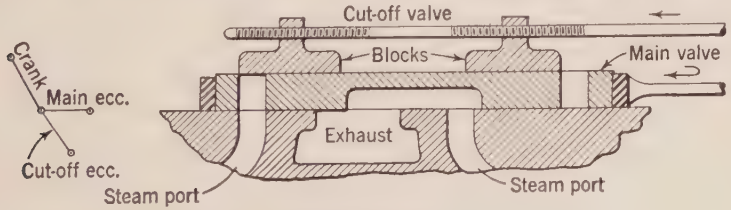


FIG. 160.—Riding cut-off valve at cut-off.

Speed changes may be accomplished by changing the tension of the coil spring. The oil cylinder and rod attached to the upper end of the

weight lever are for the purpose of dampening the action of the governor and thus preventing over-sensitiveness.

204. Riding Cut-Off Valves.—Auxiliary valves are sometimes applied to D slide-valves to control the point of cut-off independent of the other events of the stroke. In this manner one of the serious limitations of a D slide-valve may be eliminated. Illustrations of the action of a slide-valve equipped with a riding cut-off are shown for the events of admission and cut-off, by Figs. 159 and 160. The main valve is ported and normally is set to give cut-off at about $\frac{3}{4}$ the engine stroke. The main valve is driven by an eccentric rigidly attached to the engine shaft. The rider or cut-off valve is usually driven and controlled by the mechanism of a shaft governor. The sole function of the rider valve is to give earlier

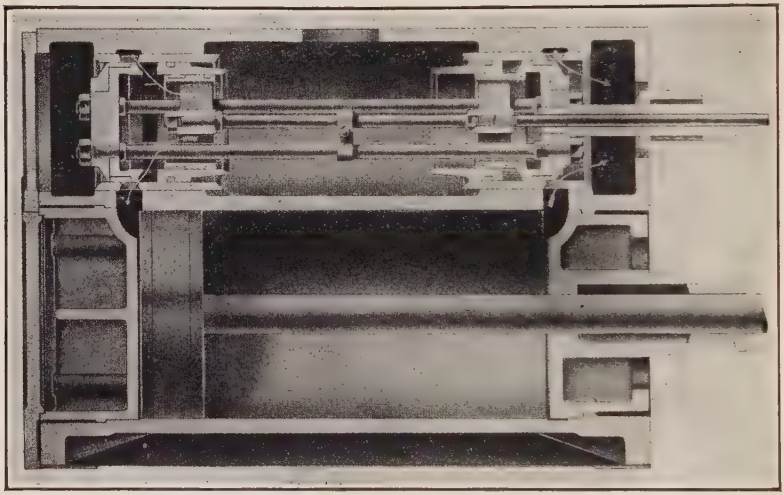


Fig. 161.—Buckeye piston riding cut-off valve.

cut-off than is possible with the main valve, without interference with the events of admission, release, and compression.

The riding cut-off principle is applied to the piston valves of the Buckeye engine, Fig. 161. The Buckeye valve consists of two concentric piston valves, the outer one of which is the main valve. The inner valve is actuated by a shaft governor through a valve rod which works inside of the valve rod moving the main valve. As is common with most piston valves the working edges of the valves of Fig. 161 are just opposite to those of the ordinary slide-valve.

205. Automatic High-Speed Engine.—A typical automatic high-speed engine is illustrated by the engine of Fig. 162. This engine is of

the center-crank type as the engine crank is midway between the main bearings of the engine frame. The engine is equipped with a Sweet balanced slide-valve and a centrifugal governor. The valve motion is

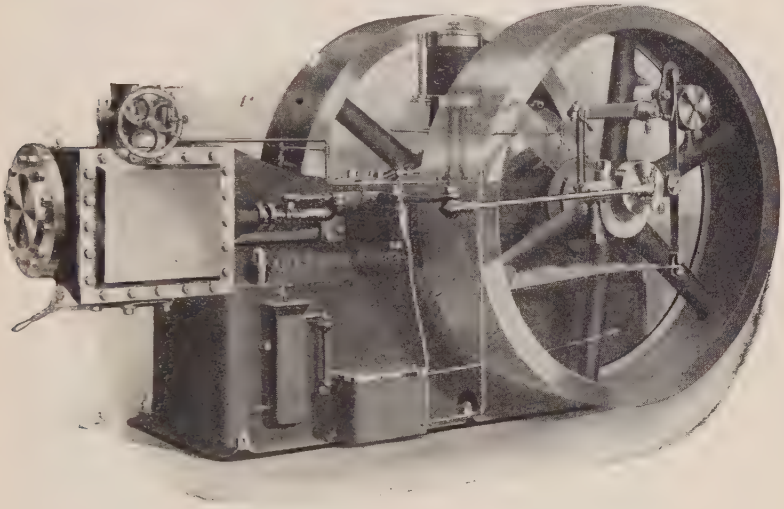


Fig. 162.—Ames automatic high-speed slide-valve engine.

transmitted from the governor parts by the eccentric rod to a horizontal rocker, which moves the valve slide by a slotted crosshead. Engines of this type may be had in sizes ranging from 30 to 400 hp.

CHAPTER XIV

MULTI-VALVE AND MULTI-EXPANSION ENGINES

206. Foreword.—Engines equipped with two or more valves have operating and other advantages that secure better thermal economy. Early cut-offs are permissible without interference with the proper distribution of the steam within the cylinder as the valves may be adjusted independently. The action of the valve at cut-off may be

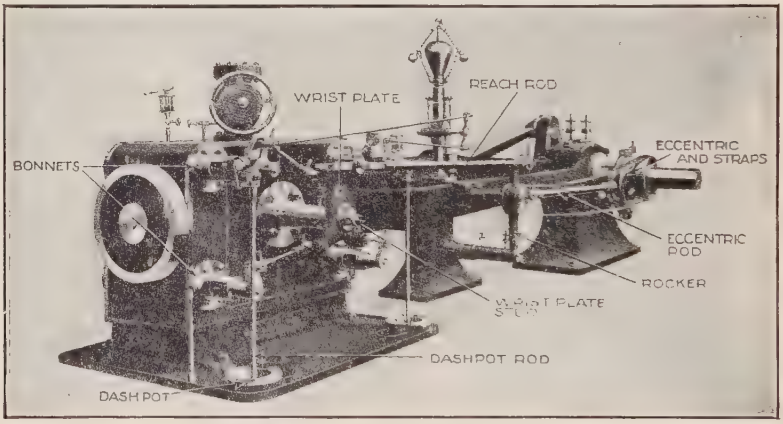


Fig. 163.—Murray Corliss steam engine.

made more rapid and wiredrawing losses due to the valve reduced. Port passages may be made shorter and the clearance of the engine reduced. The greater part of the gains in thermal efficiency secured are due to the reduction of initial condensation and wiredrawing losses.

207. The Corliss Engine.—In the middle of the nineteenth century George Corliss developed the four-valve type of engine named for him. This engine showed improved economy over anything in use prior to that time. Fig. 163 illustrates a typical single eccentric Corliss engine with a girder frame. The cylinder casting is made as shown by Fig. 164 and has four valve chambers. The two steam valves used for the admission and cut-off of the steam are located in cylindrical chambers at the top of the cylinder casting. The exhaust events of the cycle are controlled by

cylindrical exhaust valves located at the bottom of the cylinder block. Very short ports lead from the valve chambers to the cylinder bore.

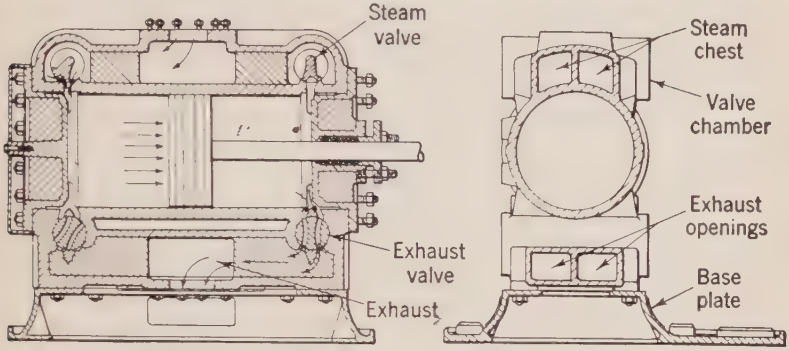


FIG. 164.—Murray Corliss engine cylinder with valves.

The steam and exhaust valves are constructed as shown by Fig. 165, and are given a semi-rotary motion by the valve gear. The exhaust valves illustrated are double ported while the steam valves are constructed to give the effects of double ports. Corliss valves are made with single, double or triple ports.

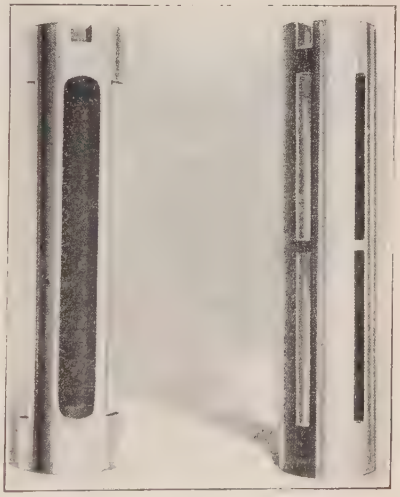


FIG. 165.—Corliss steam and exhaust valves.

With the exception of the valves and the valve gear, the essential features of a Corliss engine are those common to many steam engines. The valves of the engine of Fig. 163 are actuated by a single eccentric mounted upon the engine shaft. The eccentric motion is transmitted by the eccentric straps and eccentric rod to a vertical rocker arm pivoted at its bottom end. Motion from the upper end of the rocker is carried by the reach rod to the wrist plate mounted upon a stud fastened to the side of the cylinder casting. The wrist plate is given an oscillating motion which is used to move the valves.

Each valve is rotated by a T-headed spindle which fits into the slot in the end of the valve. Bearings for the valve spindles are located in

the supporting bonnets and steam-tight packings are provided about the valve spindles to prevent leakage of steam from the valve chambers. Each valve spindle has an arm keyed rigidly to it. The levers attached to the steam valve spindles are steam arms and those attached to the exhaust valve spindles are exhaust arms. Exhaust valve rods connect the exhaust arms and the wrist plate directly, thus the exhaust valves are opened and closed by the wrist plate. In the case of the steam valves the steam-valve rods driven by the wrist plate are attached to bell cranks or double levers, Fig. 166. The double levers are mounted upon the valve spindles, and are free to turn upon them. Each double lever carries a steam hook at the end opposite to the steam valve rod attachment. When the double lever is rotated to the proper position

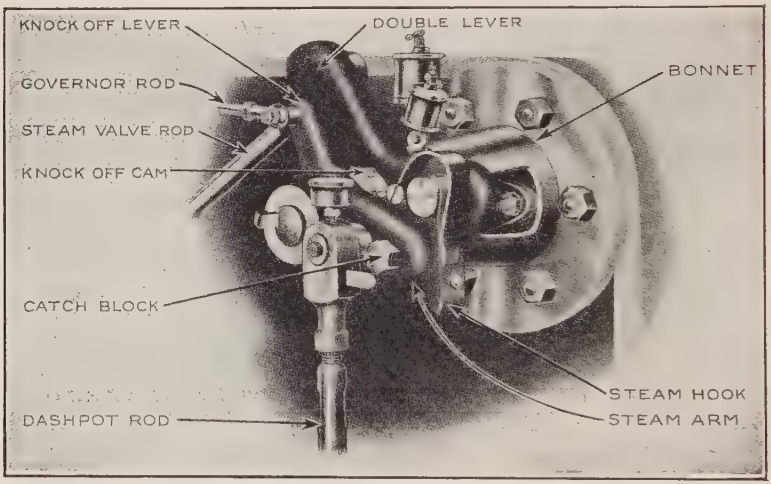


Fig. 166.—Murray Corliss steam-valve gear.

by the wrist plate the steam hook engages with a catch block fastened to the steam arm. Motion of the double lever causes the valve to be rotated to a position for admission of steam to the cylinder. Attached to the steam arms are rods which lift dash-pot plungers, Fig. 163. The dash pot plungers are lifted against the pressure of the atmosphere as a partial vacuum is produced under the plunger when it is lifted. Admission of the steam into the cylinder continues until the steam hook comes in contact with the knock-off cam mounted upon the knock-off lever, Fig. 166. When this contact is made the steam hook is disengaged from the catch block, and the valves are quickly closed by the air pressure exerted on top of the dashpot plungers. Thus very rapid and sharp cut-off is secured. The knock-off levers rotate about the axes of

the valve spindles and the positions of the knock-off lever and the knock-off cam are directly under control of the governor.

Governing is effected by a loaded pendulum governor shown by Fig. 167. The governor action is transmitted by the long governor rod to the knock-off lever at the head end of the cylinder and by the short governor rod to the knock-off lever at the crank end of the cylinder. Cut-off governing is effected and the speed regulation of the engine is generally good. Speed changes may be made by increasing or decreasing the amount of shot held in the governor weight, Fig. 167. Due to the fact that the point of cut-off is under governor control, the number of steam expansions in a single cylinder are variable, depending upon the load on the engine.

In case the governor belt breaks, the speed of the engine is brought

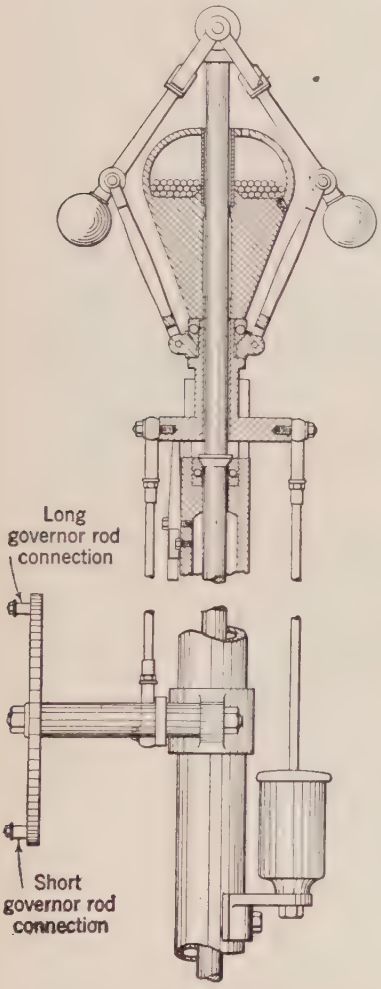


Fig. 167.—Corliss engine governor.

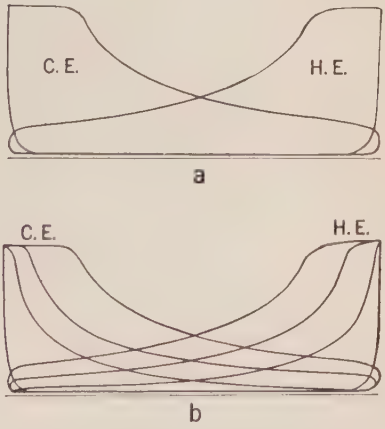


Fig. 168.—Corliss engine indicator diagrams.

under control as the governor weights cease to revolve. This control is accomplished by the governor rods shifting the knock-off levers to a position where parts known as safety cams prevent the steam hooks from engaging with the catch blocks. When this action occurs the steam valves are not opened and the engine stops.

Due to the reduced clearance volume, reduced initial condensation losses, and cut-off governing, the Corliss engine gives better heat economy than throttling-governed slide-valve engines. Typical indicator diagrams for a single eccentric Corliss engine at full and varying loads are shown by Fig. 168*a*. The effects of the governor action may be seen by Fig. 168*b*.

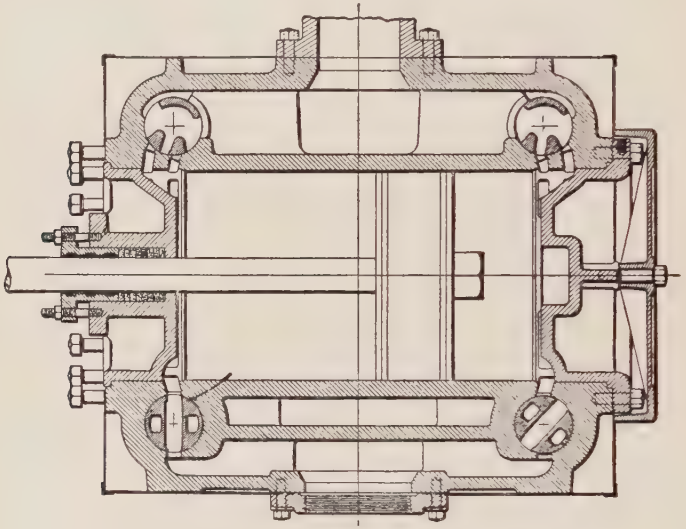


Fig. 169.—Erie Ball non-releasing Corliss engine cylinder.

208. Corliss Valve Gears.—The types of Corliss valve gears in use are:

(1) Single eccentric and double eccentric.
(2) Releasing.
(3) Non-releasing.
(4) Long range.

The single eccentric gear, of the engine of Fig. 163, cannot be used with cut-off much later than 3/8 of the stroke. If later cut-off is attempted with all valves actuated by a single wrist plate, the events of release and compression cannot be properly timed. In order to eliminate the interferences developed with late cut-offs in single eccentric engines, Corliss engines are built with two eccentrics and two wrist plates. One eccentric with its wrist plate moves the steam valves and the other eccentric with its wrist plate controls the exhaust valves. Engines equipped with two eccentrics may have cut-off as late as 7/8 of the stroke.

The engine, Fig. 163, has a *releasing gear*, in that the steam valves are disengaged from the driving gear at cut-off, and are closed by the

action of the dashpots. Engines thus equipped are not suited to rotative speeds above 100 to 125 r.p.m., as the releasing gear and the dashpots do not function properly with greater speeds.

Four-valve engines of the Corliss type suitable for higher rotative speeds are built with a *non-releasing gear* (Art. 209). Single wrist-plate engines can be given a *long range* of cut-off by the attachment of oscillating cams to the steam arms. These cams act to positively throw the steam hooks or latches out of engagement at the proper place in the

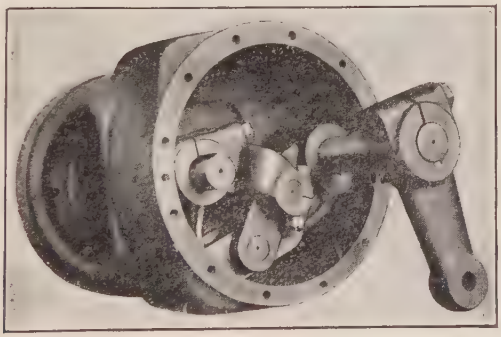


FIG. 170.—Armstrong non-detaching valve gear.

stroke and thus secure cut-off where desired. The oscillating cams are moved to the proper positions by the governor acting through the governor rods. The cam motions are derived from a separate eccentric and the point of cut-off is determined by the cut-off eccentric and the governor. The long-range gear permits higher rotative speeds, and cut-offs up to 0.8 of the stroke may be obtained by single wrist-plate engines.

209. Non-releasing Corliss Engine.—A section of a cylinder using Corliss valves and a non-releasing gear is illustrated by Fig. 169. The steam valves shown are double ported, although they may be made single or triple ported, depending upon the cylinder size.

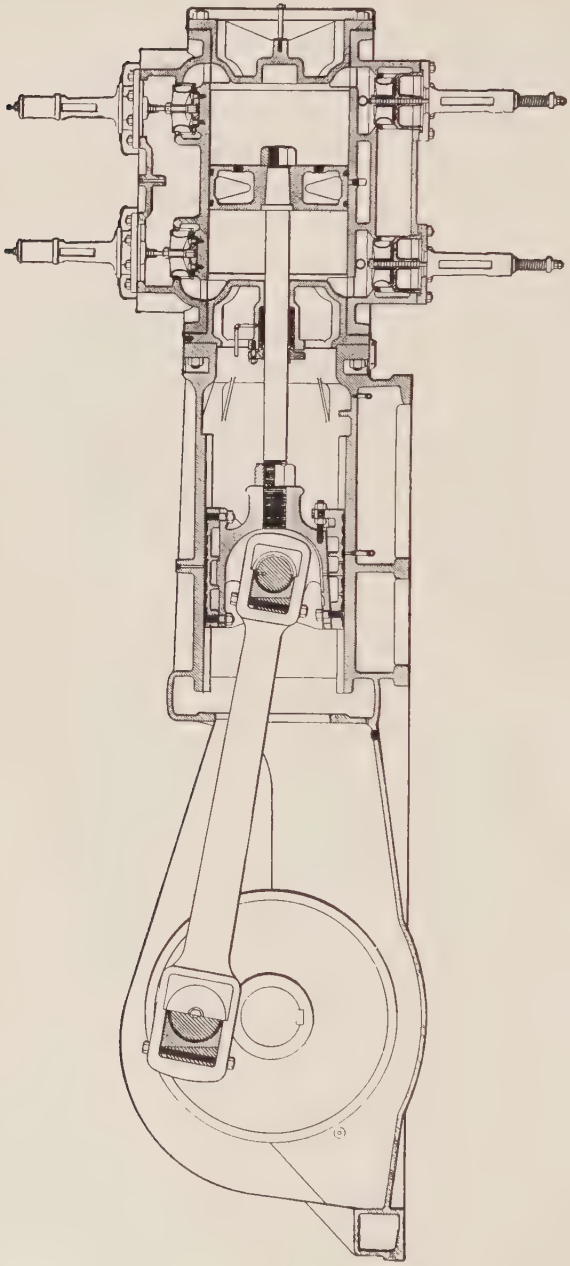


FIG. 171.—Longitudinal section of an Erie City Lentz poppet-valve steam engine.

The steam valves are actuated by links and levers enclosed within oil tight housings, Fig. 170. The motion imparted to the steam valves is almost identical with that obtained from a releasing gear with the added advantage that higher rotative speeds are permissible. The steam valves are opened and closed at the proper times and are held stationary during the remainder of the time. Small movements of the steam valves are required, and they are balanced while opening and are nearly balanced during the period of closure. The exhaust valves are driven from a wrist plate as is customary with Corliss engines.

Governing is effected by the use of a flywheel governor which changes the position of the shifting eccentric used to operate the steam valves.

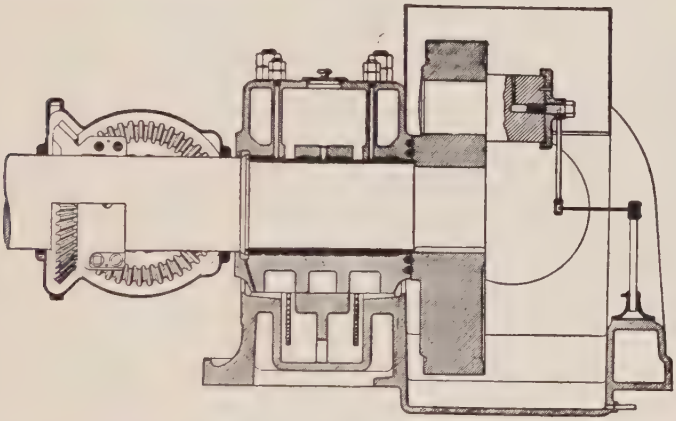


Fig. 172.—Section of the main bearing of a Lentz poppet-valve steam engine.

210. Poppet-Valve Engine.—Poppet-valve engines are a European development adapted to the use of highly superheated steam. The longitudinal section of a poppet-valve engine is shown by Fig. 171. This engine has two inlet and two exhaust valves, all of which are operated by means of eccentrics and rods from a lay-shaft driven by the bevel gears, Fig. 172. The poppet valves are of the double-seat type. More details of the valves and the operating gear are shown by Fig. 173.

The engine is governed by a combination centrifugal-inertia governor, mounted upon and driven by the lay-shaft. Speed variations of the engine cause centrifugal weights and an inertia ring within the governor housing to be shifted. The movement of the component parts within the governor housing causes the throw of the slotted eccentrics operating the steam valves to be changed. In this manner the point of cut-off is varied within the engine cylinder to suit the load conditions. Speed

changes of the engine may be effected while the engine is in operation by adjusting the tension of the governor spring. This adjustment is accomplished by the aid of a small hand wheel at the end of the lay-shaft.

The valve spindle movements are derived from the eccentric rods by the action of a bell crank cam. The cams bear on rollers until the valves are seated by the action of the valve springs. The cam contacts are disengaged as the valves seat, noise is eliminated and the engine may be operated at any reasonable speed.

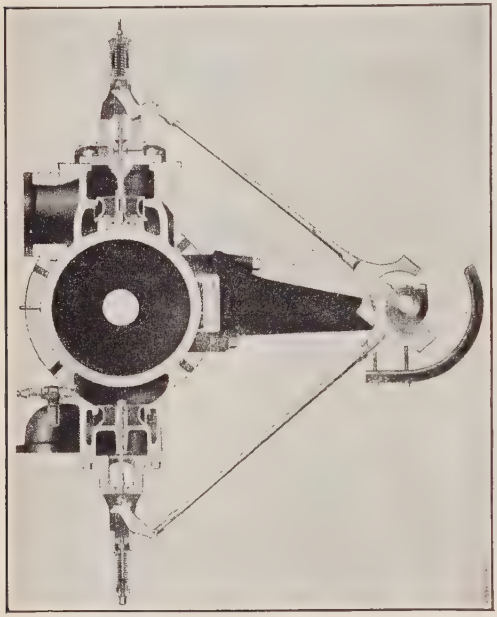


FIG. 173.—Valve gear of a Lentz poppet-valve engine.

Poppet valves permit the use of highly superheated steam as difficulties of lubrication and warping of valves are eliminated. The use of the highly superheated steam, together with reduced clearances, etc., reduce the thermal losses within the engine and good operating economy results. These engines are built both in simple and compound units.

211. Unaflow Engine.—The single-flow principle, as well as the benefits derived from its use, were explained by Art. 186. The best results, due to the application of the single-flow principle, are obtained from condensing engines having a single valve at each end of the cylinder, as the example of Fig. 174. Condensing operation is desirable for all unaflow engines as ordinarily compression of steam occurs during about 90 per cent of the return stroke. In condensing operation the compression pressure may rise to that of the initial steam, but in non-condensing operation the compression pressure, unless relieved in some manner, will exceed the initial steam pressure. Condensing unaflow engines, operating with exhaust pressures considerably below atmospheric pressure, have a reduced weight of steam entrapped within the cylinder, when the piston covers the regular exhaust ports. Consequently, as the weight of steam compressed is not large, the pressures of compression do

not become excessive. Engines operating non-condensing have a greater weight of steam held within the cylinder during compression and the pressures become excessive with a loss of engine power and steam economy. Non-condensing engines are often fitted with auxiliary clearances, steam ports or automatically or mechanically operated relief valves to reduce the compression pressure. These devices permit the development of more power than could be obtained without their aid in non-condensing operation, but the use of the expedients to relieve compression partially nullify the thermal gains to be secured by application of the unaflow principle. However, the non-condensing unaflow engine

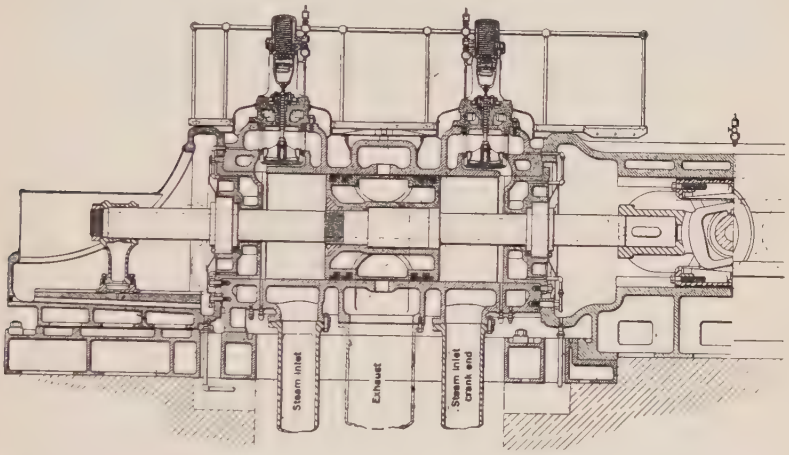


Fig. 174.—Sectional elevation of a Mesta unaflow engine cylinder. (Stumpf patents.)

is able to show materially lower water rates than counterflow engines operating under comparable conditions.

Figure 175 shows the assembled view of the crank side of a unaflow engine with exhaust valves and tail rod. The valve arrangement of the engine is shown by Fig. 176. Two poppet inlet-valves are provided for the admission of steam at the top of the cylinder. The usual exhaust ports, opened and closed by the engine piston, are located at the middle of the cylinder barrel. A part of the steam entrapped in the cylinder after the regular exhaust ports are closed by the piston is released through the mechanically operated poppet exhaust valves at the bottom of the cylinder. With the exception of the two exhaust valves, the construction is similar to that of any two-valve unaflow engine.

Fig. 177 illustrates a cylinder relief valve which is used by Elliott unaflow engines built for condensing operation. This valve is used when the vacuum fails and the compression pressures become excessive.

The small inside valve is hand-operated and serves as a cylinder drain. The larger valve is a spring-loaded valve which can be adjusted by an

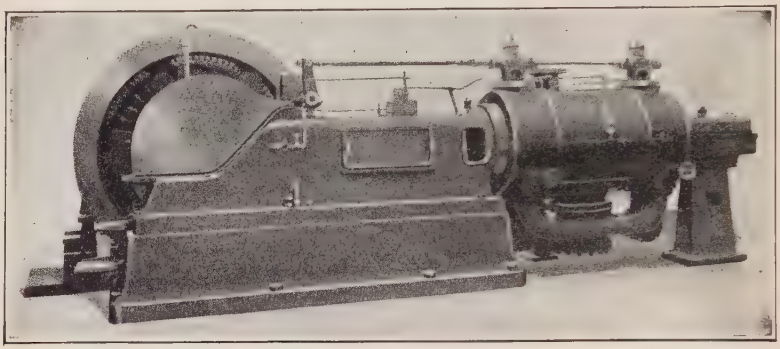


Fig. 175.—Elliott simple unaflow engine, with auxiliary exhaust valves.

outside nut. When the compression pressure becomes excessive the spring-loaded valve opens under the pressure and permits compressed

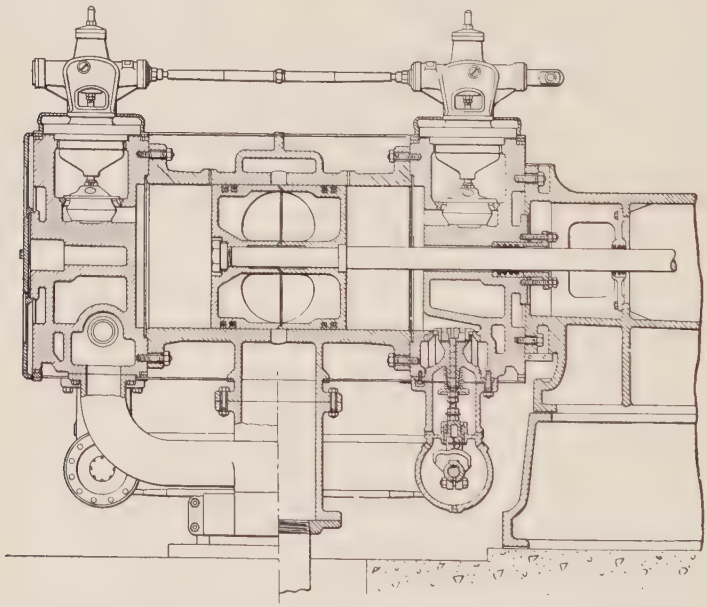


Fig. 176.—Sectional elevation of an Elliott simple unaflow engine, showing exhaust valve and operating mechanism.

steam to flow into the exhaust. When non-condensing operation is necessary the large outside valve may be opened by means of the hand wheel, and a port uncovered between the cylinder and a clearance pocket.

212. Economy of the Unaflow Engine.—The improved thermal efficiency of the unaflow engine is secured primarily by the reduction of condensation losses, through the use of the single direction of steam flow and the use of jacketed cylinder heads and barrels. Turbulent and cooled steam does not reduce the temperature of the cylinder walls and ports at the inlet end of the cylinder, and a minimum temperature difference is always maintained between that of the steam and that of the confining walls. The exhaust passages around the cylinder barrel are filled with steam which, in a way, serves as a cylinder jacket. The cylinder clearances are reduced as well as steam leakage through the valves. The engine is adapted to high initial steam pressures, temperatures, and high rotative speeds, and is not handicapped by difficulties of valve lubrication and valve warping.

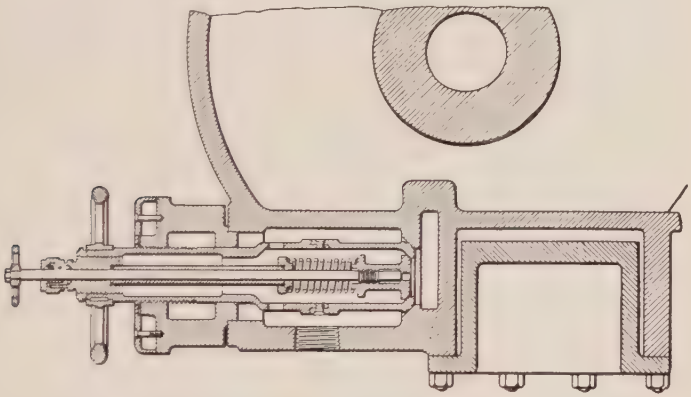


FIG. 177.—Elliott unaflow engine relief valve, for condensing operation.

Unaflow engines have smaller variations of the amount of steam consumed per horsepower-hour with variation of loads than compound engines show, Fig. 178. As a matter of fact, most compound engines show good economy at one load only. Comparative indicator diagrams of a single-cylinder unaflow engine and a compound engine, operating under identical conditions, are shown by Fig. 179. The diagrams of the two cylinders of the compound engine have been combined into a single diagram, and the theoretical cards for both the unaflow and the compound engine have been constructed. The cross-sectioned areas represent the lost work areas in each case.

Unaflow engines are more economical in many cases than small turbines operating under comparable conditions, and in recent years have largely superseded the compound steam engine where variable loads imposed in engine operation occur.

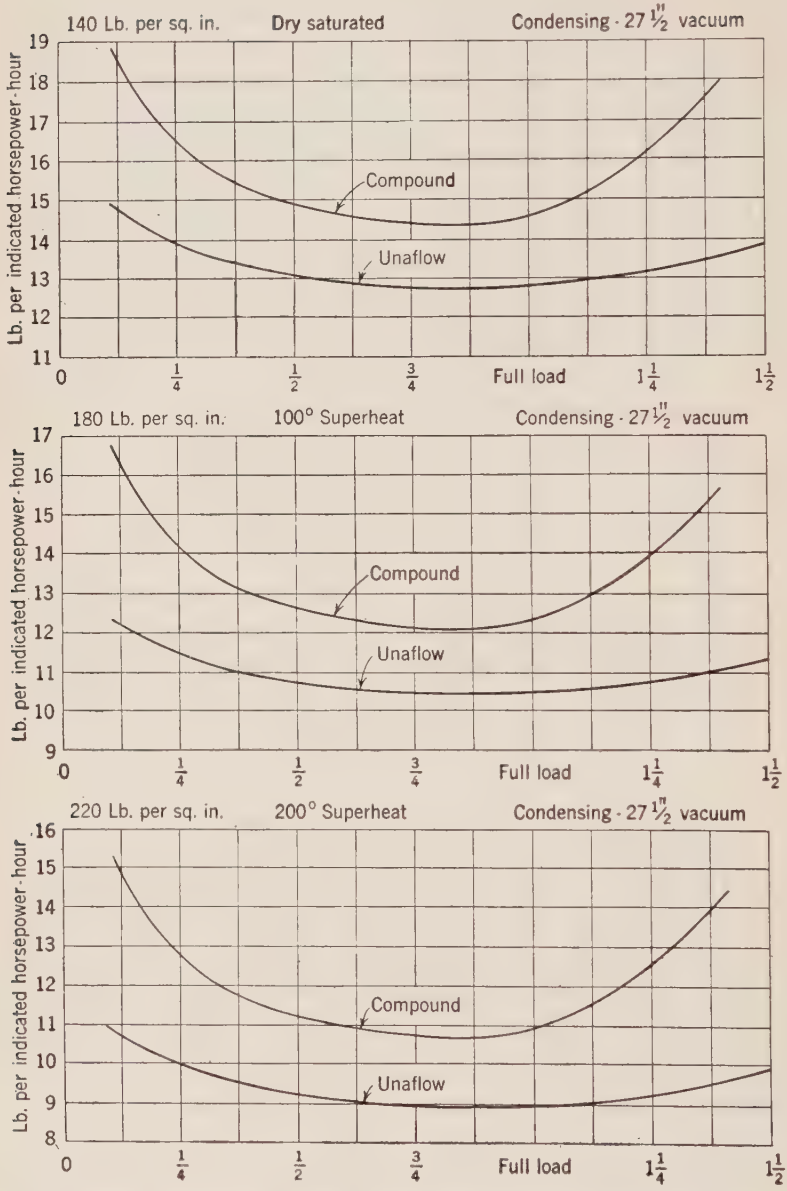
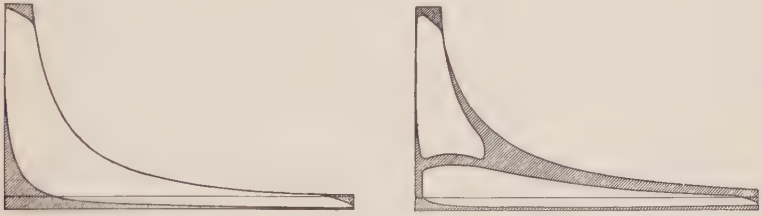


FIG. 178.—Steam consumption curves. Mesta single-cylinder unaflow, and a compound counterflow engine.

213. Arrangement of Compound Engines.—Engines with two or more cylinders for the successive expansion of steam are built so that the classifications of Art. 156, Chapter XI, are applicable. In addition to the groupings there given, further differentiations of the cylinder locations may be made:

(1) Tandem-compound, cylinder longitudinal axes on the same center line.
(2) Cross-compound, cylinder longitudinal axes on parallel center lines. Engine cranks at opposite ends of the crankshaft.
(3) Duplex-compound, cylinder longitudinal axes parallel. Both cylinders served by the same crosshead and connecting rod.
(4) Angle-compound, cylinder longitudinal axes at right angles.

The tandem and cross-compound arrangements are those most commonly used.



Unaflow engine. Compound counterflow engine.

FIG. 179.—Comparison of unaflow and compound-engine indicator diagrams.

214. Tandem-Compound Engines.—Engines with the cylinders in tandem have the pistons of the two cylinders mounted upon a single piston rod, Fig. 180. The low-pressure cylinder is usually placed nearest the crankshaft. This location restricts access to the cylinder. Tandem-compound engines require a smaller amount of floor space than those of the cross-compound type.

Tandem-compound engines may be built with valves controlling cut-off of the steam in the high-pressure cylinder only, or with valves controlling cut-off in both the high- and low-pressure cylinders. In the case where a cut-off valve is provided for the high-pressure cylinder only, the flow of steam is as follows: During admission and expansion of steam in the head end of the high-pressure cylinder, exhaust steam flows from the crank end of the high-pressure cylinder into the head end of the low-pressure cylinder. During admission and expansion of the steam in the crank end of the high-pressure cylinder, exhaust steam flows from the head end of the high-pressure cylinder into the crank end of the low-pressure cylinder. In each case, cut-off takes place in the low-pressure cylinder

at the end of the stroke, and disposal of the exhaust from the low-pressure cylinder is made in the usual manner. Connecting pipes are

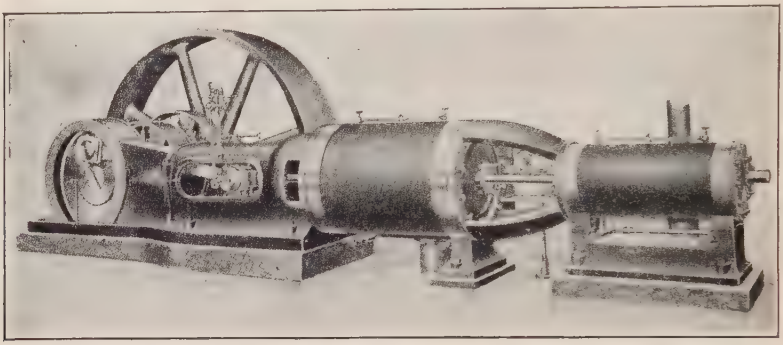


Fig. 180.—Buckeye tandem-compound steam engine.

provided between the proper cylinder ends. Such an engine is of the "Woolf" type.

Valves may be provided to give cut-off in both the high- and low-pressure cylinders, as shown by the Buckeye riding cut-off valves of Fig. 181. This design necessitates the use of a *receiver*.

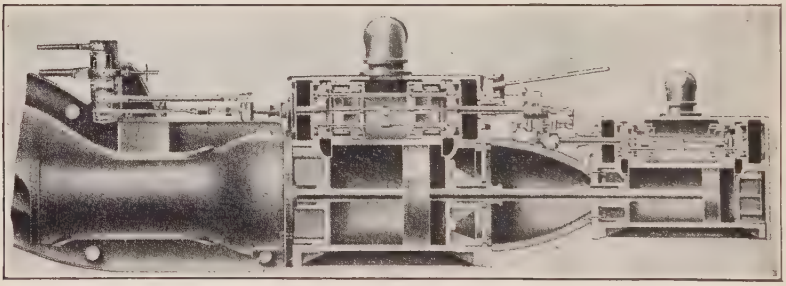


Fig. 181.—Section of Buckeye tandem-compound engine cylinders.

215. Engine Receivers.—Storage space for exhaust steam from the high-pressure cylinder is necessary, and is secured by the use of receivers for the following conditions: (1) When the exhaust of steam from the high-pressure cylinder cannot pass continuously to the low-pressure cylinder. (2) Engines having the low-pressure crank ahead of the high-pressure crank and requiring admission of steam to the low-pressure cylinder, when the high-pressure cylinder is not exhausting. Receivers may consist merely of enlarged pipes between the cylinders or they may be built of steel plates riveted together. When reheating of the steam is desired between cylinders, coils for high-pressure steam are inserted within the receiver.

216. Cross-Compound Engine.—An engine of this type usually has overhung cranks at the ends of the crankshaft with the engine flywheel mounted between the main bearings of the engine. Ordinarily, the low-pressure cylinder crank is set 90 deg. ahead of the high-pressure crank, in the direction of rotation. This arrangement permits a more uniform turning effort on the crankshaft, and necessitates the use of a receiver. A typical cross-compound engine having high- and low-pressure cylinders of the four-valve type is illustrated by Fig. 182. Engines of this type require considerable floor space, but the parts may be made lighter.

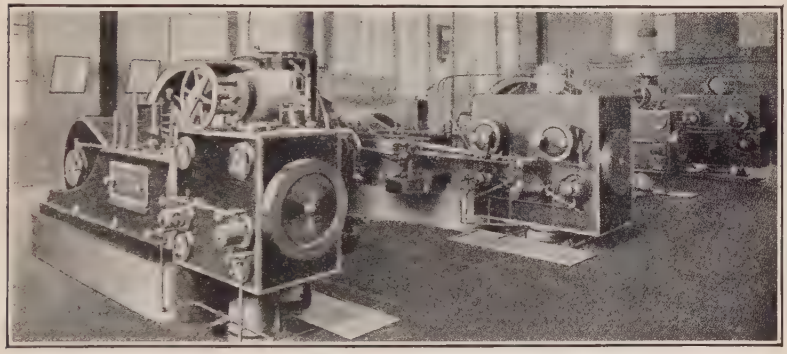


FIG. 182.—Erie Ball cross-compound engine with non-detaching valve gear.

217. Compound Engine Cylinder Ratio.—The ratio of the piston displacement of the low-pressure cylinder to that of the high-pressure cylinder is the cylinder ratio. For any engine the ratio is

$$R = \frac{\frac{\pi}{4} D_2^2 l_2}{\frac{\pi}{4} D_1^2 l_1}$$

$$= \frac{D_2^2 l_2}{D_1^2 l_1} \quad \ldots\ldots\ldots \quad (134)$$

in which D_1 = diameter of high-pressure cylinder, in.
D_2 = diameter of low-pressure cylinder, in.
l_1 = length of stroke high-pressure cylinder, in.
l_2 = length of stroke low-pressure cylinder, in.

When the strokes of the high- and the low-pressure cylinders are equal, as often is the case

$$R = \frac{D_2^2}{D_1^2} \quad \ldots\ldots\ldots \quad (135)$$

The cylinder ratio should be such that a small drop in steam pressure occurs between cylinders when the power developed by them is equally divided.

218. Ratio of Expansion.—The ratio of the total volume of the low-pressure cylinder to that of the high-pressure cylinder at the point of cut-off is the actual expansion ratio r. In each cylinder the volume under consideration is the piston displacement plus the clearance volume of the cylinder end under consideration.

219. Power of a Compound Engine.—Engines operating without bleeding of steam from the receivers between cylinders, discharge in the low-pressure exhaust practically all of the steam supplied to the high-pressure cylinder. The calculation of the theoretical power of a compound engine is made on the basis that all of the work may be done in a single cylinder having the dimensions of the low-pressure cylinder. The number of steam expansions in the low-pressure cylinder is considered as being the same as the total of the expansions of both the high- and the low-pressure cylinders. On this basis the theoretical mean effective pressure of an equivalent simple engine may be calculated by the use of equation 120, Chapter XI. As the probable m.e.p. is always less than the theoretical m.e.p., it is necessary to apply a diagram factor to the m.e.p. thus found. Diagram factors for compound engines range from 0.5 to 0.8, depending upon the type of the engine and the valve gear used. Any type of valve and valve gear may be used, such as slide, Corliss, or poppet. Having found the probable mean effective pressure, the probable power of the engine may be calculated by the use of equation 122, Chapter XI.

The actual power of a compound or multi-expansion engine is the sum of the powers developed simultaneously in all of the cylinders.

220. Governing.—Compound engines, except some used for driving air compressors, are governed by varying the point of cut-off. The governors used are of the centrifugal or inertia shaft type, or pendulum-loaded governors for engines with Corliss releasing valve-gears.

Cut-off governing may change only the point of cut-off in the high-pressure cylinder or the point of cut-off may be varied in both the high-pressure and the low-pressure cylinders. The total power developed by a compound engine is dependent upon the point of cut-off in the high-pressure cylinder. The distribution of work or power between the cylinders is adjusted by the point of cut-off in the low-pressure cylinder. Making the cut-off earlier in the low-pressure cylinder increases the power of the low-pressure cylinder, and decreases the power of the high-pressure cylinder. The effect is caused by the increased receiver pressure, which causes the high-pressure piston to work against a greater back pressure

and the low-pressure cylinder to work with a higher initial pressure. Late cut-off in the low-pressure cylinder produces the opposite effect as more work is done in the high-pressure cylinder and less in the low-pressure cylinder.

221. Triple- and Quadruple-Expansion Engines.—Engines expanding steam in four successive cylinders or stages are seldom used at the present time. Triple-expansion engines expand the steam in three or more cylinders. It is not uncommon to find triple-expansion engines fitted with five cylinders; one for the partial expansion of high-pressure steam; two intermediate cylinders; and two low-pressure cylinders.

The common applications of triple-expansion engines in recent years have been only to the propulsion of ships and operation of pumping units for water supplies. The recent tendency has been to discard the triple-expansion engine in both of these services. Modern ships are driven either by turbine generators through electric motors, by turbines through speed-reduction gears, or by internal-combustion engines of the Diesel type. Although the triple-expansion engine has shown remarkable economy in pumping plants, the general tendency is to use centrifugal pumps, which are driven by electric motors. While the centrifugal pump thus driven does not show as good performance in the matter of economy, the amount of floor space required and the greater simplicity of the pumping unit are factors favoring the discard of the triple-expansion engine.

However, there is under consideration, at this time (December, 1928), an industrial installation of two 5000-hp. vertical triple-expansion steam engines. These engines are to be used in the Lockland, Ohio, plant of the Philip Carey Mfg. Co. The engines are to use steam at a throttle pressure of 1400 lb. per sq. in. gage, and at an initial temperature of 800 deg. fahr. The engines are to exhaust against a back pressure of 60 lb. per sq. in. gage; and they are to be direct-connected to 3-phase alternators.

CHAPTER XV

STEAM TURBINES

222. Development.—Shortly after the beginning of the twentieth century the steam turbine began to be applied to commercial installations and to-day has supplanted the steam engine for large electric generating stations. The steam turbine can be built in units of much larger capacity than is practicable with reciprocating engines. However, the reciprocating engine is being used for certain classes of service such as hoisting engines, piston compressors, small industrial plants, belt drives, etc. The development of the turbine has been remarkably rapid, but at the same time improvements in other central station equipment have been made, especially in boiler-plant construction and operation, condensing apparatus and electric generator design. Single-cylinder units have been built in various sizes ranging from 1 hp. to large turbine-generators of 75,000 kw. rated capacity. Multiple-cylinder units developing 100,000 kw. have been installed, and a 208,000-kw. triple-compound unit is being installed (1929) in the State Line Generating Station near Hammond, Indiana. One manufacturer has offered to build a multi-cylinder unit in excess of 300,000 kw.

The steam in a turbine is continuously directed from nozzles or guide passages against semicircular blades (or buckets) uniformly spaced on the periphery of an enclosed rotor. The reciprocating steam engine utilizes the available energy of the steam by the action of a static pressure against the piston. In a steam turbine the energy transformation is also accomplished by a pressure action on the blades of the rotor but the pressure is dynamical, being due to a change of momentum of the steam in its passage through the blade channels. As expansion proceeds in the steam engine, the heat energy is directly transformed into work on the piston. In the turbine there is a double transformation of energy; heat is first converted into kinetic energy by the expansion of the steam in the nozzles and this kinetic energy is then transformed into mechanical work on the rotor blades.

Compared with reciprocating engines of the same capacity and operating conditions, steam turbines occupy less floor space, require smaller and cheaper foundations, deliver oil-free exhaust steam, operate

with highly superheated steam without lubrication difficulties, and in general, show about the same heat economy and thermal efficiency. Other advantages are uniform velocity of rotation, higher rotative speeds, smaller initial cost, and less attendance required. The steam turbine is primarily adapted to the driving of high-speed machinery such as electric generators, centrifugal pumps, rotary compressors and fans, but is not applicable to direct drives where relatively low speeds are desired or where a large starting torque is necessary. The high speed of the turbine may be reduced by gears, but this arrangement increases the size, weight, and cost of the unit and decreases the mechanical efficiency.

223. Classification of Steam Turbines.—As in the case of reciprocating engines, steam turbines may be classified under the headings of "Simple" and "Compound," but a more general classification is usually adopted. Another classification, in accordance with the conditions of operation of the steam on the rotor blades, is: (1) Impulse, (2) Pure Reaction, and (3) Reaction. In all important commercial types of steam turbines, the blades are moved by both the impulse and the reaction of impinging steam jets issuing from nozzles or guide passages.

Fig. 183 shows a nozzle and rotor in which the steam in its passage through the blades flows in one direction only, the curvature of the blades being less than 90 deg. If the rotor were stationary, the steam would leave the blades in a direction nearly parallel to the shaft. Thus the only effective force for moving the blades is a "forward" pressure or *impulse* of the steam jet leaving the nozzle N. In Fig. 184 the blades have a curvature of nearly 180 deg. so that the direction of flow of the steam is reversed. The blade is acted upon by two simultaneous forces, both tending to move the rotor. When the steam jet leaves the nozzle N and strikes the blade, a forward or impulse force tends to produce motion, and as the steam leaves the blade, a backward force or *reaction*, approximately equal to the impulse, exists. If the steam has the same velocity and density, and frictional losses are neglected, the pressure exerted on the rotor in Fig. 184 will be about twice as great as on the rotor shown in Fig. 183. With the impulse type of turbine only a part of the blades are filled with steam at any instant.

In the reaction type of turbine, the steam flows through guide passages in a stationary ring (stator) S, Fig. 185, and then through blades around the periphery of the rotor R. The guides and blades are filled with steam throughout the entire circumference, the steam being partially expanded in the stator, and the blades are so made that part of the expansion occurs in the rotor. In Figs. 183 and 184, all the expansion is in the nozzles with no expansion in the blades. The reaction

type differs from the impulse type shown in Fig. 184 chiefly in the blade section *B*, shown at the top of the drawings.

In an *impulse* turbine, Fig. 184, the entire steam expansion takes place in the fixed nozzle passages. As there is no expansion in the moving passages between the rotor blades, the steam pressure is the same at the inlet and outlet edges of these blades. There are no stationary nozzle passages in the *pure reaction* turbine and the expansion takes place wholly in nozzle passages at the circumference of a rotor. In the *reaction* type, expansion takes place in both fixed and moving nozzle passages, and the pressure at entrance to the rotor blades is, therefore, greater than at exit. The name " reaction " employed to denote this type of

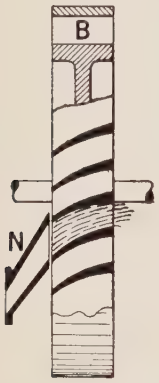


Fig. 183.—Impulse turbine, single-flow blades.

Fig. 184.—Impulse turbine, reversed-flow blades.

Fig. 185.—Reaction turbine.

turbine is not quite correct as both impulse and reaction forces are present. However, the term reaction has come into general use and will be used in this chapter to denote this type of steam turbine, Fig. 185.

224. Nozzle Design.—In the preceding articles the statement was made that heat is converted into kinetic energy when steam flows through a nozzle. An ideal impulse turbine could absorb all the kinetic energy of the steam jet and make this energy available in the form of useful power at the turbine shaft. The ideal turbine is, therefore, of interest in studying the velocity which a jet will acquire, the behavior of the steam, and the dimensions of the nozzle. To simplify matters, the assumption will be made that the steam neither receives heat from nor gives heat to other bodies during its passage through the (frictionless) nozzle, in other words, that the jet is formed under *adiabatic* conditions. Assume steam flowing through a nozzle of any form, from a

region of relatively high pressure to one of low pressure. Each element of the stream expands and the work done in expanding gives energy of motion to the element in front of it. The whole stream, therefore, acquires velocity in the process and also increases in volume. Let p_1 and p_2, Fig. 186, be imaginary partitions, across which the steam flows, taken at right angles to the direction of the stream lines. Let p_1 be the pressure of the steam at the entrance to the nozzle, V_1 the velocity, and v_1 the volume of 1 lb. of steam entering the nozzle. Similarly, let p_2, V_2 and v_2 be the pressure, velocity and volume of 1 lb. of steam at exit of the nozzle. As a result of the expansion during the drop in pressure from p_1 to p_2 the velocity increases from V_1 to V_2 and the increase of kinetic energy is equal to $\frac{V_2^2 - V_1^2}{2g}$ per pound of steam.

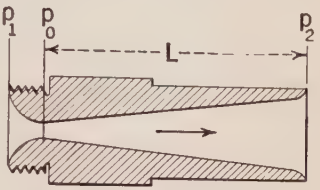


FIG. 186.—Typical turbine nozzle.

Each pound of steam that enters the space between p_1 and p_2 has work done upon it, amounting to p_1v_1, by the steam behind. In passing out of this space at p_2, work amounting to p_2v_2 is done on the steam in front of the jet. In flowing from p_1 to p_2, internal energy amounting to $U_1 - U_2$ is lost. Hence, by the principle of the conservation of energy, since by assumption no heat is taken in or given out,

$$\frac{V_2^2 - V_1^2}{2g} = U_1 - U_2 + p_1v_1 - p_2v_2 \quad . \quad . \quad . \quad . \quad (136)$$

But $U_1 + p_1v_1$ is H_1, the total heat at p_1; $U_2 + p_2v_2$ is H_2, the total heat at p_2; and the above equation may be written

$$\frac{V_2^2 - V_1^2}{2g} = H_1 - H_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (137)$$

The gain in kinetic energy is, therefore, equal to the loss of total heat or what is commonly called the "heat drop." In many practical cases the initial velocity is negligibly small, so that

$$\frac{V^2}{2g} = H_1 - H_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (138)$$

where V is the velocity in feet per second acquired as a consequence of the heat drop. In the above equations H_1 and H_2 have been treated

as though expressed in work units; when expressed in heat units they must be multiplied by the mechanical equivalent J. Since $g = 32.17$,

$$V^2 = 778.6 \times 64.34\,(H_1 - H_2)$$
$$V = 223.8\sqrt{H_1 - H_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (139)$$

The cross-sectional area of a turbine nozzle at any point n may be computed by

$$A_n = \frac{Wv_n}{V_n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (140)$$

where A_n = area, sq. ft.
W = weight of steam flowing, lb. per sec.
v_n = specific volume of steam at pressure p_n, cu. ft. per lb.
V_n = velocity of steam at point n, ft. per sec., as determined from equation 139.

Thus the area of the nozzle at any point depends upon the weight, volume and velocity of steam flowing. The assumption is, that the flow is adiabatic and frictionless, hence the entropy of the steam at any point in the nozzle is equal to the initial entropy. The design of a turbine nozzle consists simply in choosing sections so that the desired amount of steam may flow with the desired pressure drop, as the velocity is determined by that pressure drop. This is done by working in terms of 1 lb. of steam, since equations, steam tables and charts generally give values for one pound. The cross-sectional area of a nozzle at any point for passing a number of pounds of steam may be determined by multiplying the area required for one pound by the number of pounds flowing.

The shape of the entrance end of a nozzle is generally rounded by any convenient curve, as shown in Fig. 186. Experiments and mathematical determinations indicate that the steam pressure p_0 at the throat of a nozzle falls to about 0.58 of the initial absolute pressure in the case of saturated steam, and to 0.55 for superheated steam. For back pressures less than 0.58 of the initial, the nozzle must first converge from inlet to throat and then diverge from throat to exit in order to obtain a maximum velocity and provide for the increase in volume during expansion of the steam. Fig. 186 shows a section through a theoretically proportioned expanding nozzle, in which $p_0 = 0.58p_1$ and $p_2 < 0.58p_1$. To prevent friction, eddies, and velocity losses the divergence of the walls of non-condensing nozzles is about 3 deg. from the axis of the nozzle. Condensing nozzles for high vacua may have a divergence of as much as

8 deg. The total angle of divergence, therefore, varies from 6 to 15 deg. with an average value of 10 deg. For an angle of divergence of about 10 deg., the length L of the nozzle may be approximated by the equation

$$L = \sqrt{15A_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (141)$$

in which L = length of nozzle from throat to exit, in.
A_0 = area at the throat, sq. in.

The cross-sectional shape of nozzles may be either circular, elliptical, square, or rectangular with rounded corners. Experimental work indicates that the efficiency of a nozzle is practically unchanged by the shape of the cross-section when the following conditions exist. The internal surfaces must be smooth and the nozzle proportions made to give the correct ratio of the area at the throat to that at the exit end. The nozzle must not be less than, nor more than, twice the length as calculated by equation 141. Using a properly proportioned nozzle with straight sides from throat to exit, the efficiency as measured by the velocity is about 96 to 97 per cent, corresponding to an equivalent energy efficiency of 92 to 94 per cent.

Example.—(*a*) Design a theoretically perfect nozzle to pass one pound of steam per second; the steam being initially dry and saturated at a pressure of 150 lb. per sq. in. abs. and the final pressure being 14.7 lb. per sq. in. abs. (*b*) If the turbine blades could be made to transform all of the issuing jet velocity into useful work, find the horsepower imparted to the rotor. (*c*) What is the theoretical steam consumption (water rate) in pounds per horsepower per hour?

Solution.—(*a*) From steam tables, pages 33 and 34, and the heat-entropy diagram, page 47, the following values are obtained:

$p_1 = 150$ lb.	$x_1 = 100$ per cent	$s''_1 = 1.57$
$p_2 = 14.7$ lb.	$x_2 = 86.9$ per cent	$s''_2 = 1.57$
$p_0 = 87$ lb.	$x_0 = 96.0$ per cent	$s''_0 = 1.57$

$v_1 = 3.02$ cu. ft.	$H_1 = 1194.7$ B.t.u.
$v_2 = 23.3$ cu. ft.	$H_2 = 1024.4$ B.t.u.
$v_0 = 4.86$ cu. ft.	$H_0 = 1150.0$ B.t.u.

From equation 139,

$$V_2 = 223.8\sqrt{1194.7 - 1024.4} = 2920 \text{ ft. per sec.}$$

and

$$V_0 = 223.8\sqrt{1194.7 - 1150.0} = 1495 \text{ ft. per sec.}$$

Using equation 140,

$$A_2 = \frac{1 \times 23.3}{2920} \times 144 = 1.15 \text{ sq. in.}$$

and

$$A_0 = \frac{1 \times 4.86}{1495} \times 144 = 0.468 \text{ sq. in.}$$

From equation 141,

$$L = \sqrt{15 \times 0.468} = 2.65 \text{ in.}$$

The entrance end of the nozzle will be rounded as shown in Fig. 186, the throat diameter is 0.76 in., the diameter at the nozzle exit is 1.21 in., and the length of the nozzle from throat to exit is $2\frac{5}{8}$ in.

(*b*) K.E. $= \frac{WV^2}{2g} = \frac{1 \times (2920)^2}{2 \times 32.17} = 132{,}600$ ft.-lb. per sec.

Therefore Hp. $= \frac{132{,}600}{550} = 241.$

(*c*) Steam per hp. per hr $= \frac{3600}{241} = 15$ lb.

The exit end of a turbine nozzle is cut off obliquely, as shown in Figs. 183 and 184, in accordance with the angle made by the nozzle axis with the plane of blade rotation. This inclination is determined as a compromise between conflicting considerations. Higher theoretical efficiency can be obtained by using small nozzle entrance angles, as the blade action will be more efficient. Under these conditions a long nozzle is required and the frictional loss is greatly increased, the total deviation of the steam in the blading is increased, causing additional frictional losses, and the diagonally cut exit end of the nozzle is so wide that it fails to direct the steam into the blading properly. If this nozzle angle becomes too great, the blading loss becomes large, as the steam jet strikes the back of the blades at the entrance. Nozzle entrance angles used in practice range from 10 to 30 deg.—15 to 20 deg. being common.

225. Single-Stage Impulse Turbine.—This is the simplest type of steam turbine and consists principally of a single wheel revolving in a casing, the latter fitted with one or more nozzles. The De Laval "Class A" is the oldest turbine of this type; the essential elements of which are shown in Fig. 187. The steam is expanded from the initial to the exhaust pressure in nozzle N, and the kinetic energy of the jet is absorbed by a single row of blades B fastened to the periphery of wheel W. The blades deflect the jet as shown and absorb the greater part of the kinetic energy, so that the steam flows away from the wheel with a

low velocity. Many turbines of this type are in use but they are no longer manufactured because of the high rotative speeds, 10,000 to 30,000 r.p.m. For driving lower speed machines it is necessary to use reduction gearing between the turbine shaft and the driven machine.

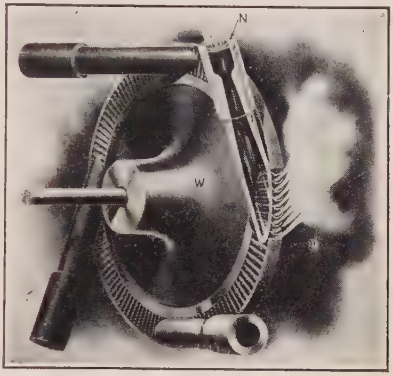


Fig. 187.—Principle of the impulse turbine.

As many nozzles are used as may be required to make available the amount of energy desired at full load, and provision is made for shutting off manually one or more nozzles as conditions may warrant. Governing for variations in load is effected by a throttling governor placed in the steam supply line near the turbine. By reducing the pressure of the steam admitted to the nozzles at light loads, the weight of steam discharged upon the blades is less than it is at a higher pressure, and correspondingly less energy is given to the turbine wheel to maintain a constant speed. The blades are made with bulb shanks, which are forced into slots milled in the circumference of the wheel. The outer ends of the blades have projections which fit closely together and are peened so as to form a continuous ring. Thus, if an individual blade becomes damaged, it may be removed and another put in its place. This type of turbine has been built in 15 sizes from 7 to 700 hp., with peripheral speeds as high as 1300 ft. per sec.

226. Action of Steam on Impulse Blades.—In an ideal impulse turbine the steam jet issuing from a nozzle would be brought to rest in the blades and would thus give up all of its kinetic energy. In the actual turbine it is impossible to bring the jet to rest, due to the limitations of practical designs. There is, therefore, always a loss in a turbine because of the residual or terminal velocity of the steam leaving the wheel.

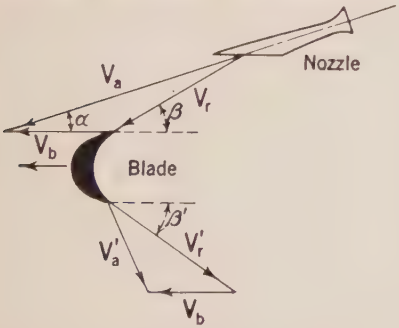


Fig. 188.—Velocity diagram for impulse turbine blades.

Fig. 188 shows a section of a blade in a single-stage impulse turbine. If the minimum eddy loss at entrance is to be obtained at the point at which the steam jet enters the blade, the jet must enter along a tangent

to the inside curve at the entrance edge. This direction is shown by the line V_r in the figure and if the blade were stationary, the steam jet would move as shown by V_r. Since the blade moves ahead and away from the jet, the steam must be directed as indicated by V_a in order to strike the blade in the direction indicated by V_r.

In the case of steam impinging on a blade as shown in Fig. 188, the nozzle directs the jet against the blade with absolute velocity V_a. A nozzle angle α (about 20 deg. with the plane of the wheel) is used. Since the blade moves at a velocity of V_b, the velocity V_r of the steam jet relative to the blade is the geometrical resultant of V_a and V_b. For convenience in construction the entrance angle β and exit angle β' are made equal. Neglecting frictional losses in the blade, the relative exit velocity $V'_r = V_r$ and the resultant of V'_r and V_b is the absolute exit velocity V'_a. Obviously, the energy removed from steam passing over the blade is

$$\frac{WV_a^2}{2g} - \frac{W(V'_a)^2}{2g}$$

If the blade were theoretically perfect the absolute velocity V'_a would be zero and all of the kinetic energy would then be removed. This is practically impossible in an actual turbine and there is always a loss due to the residual velocity V'_a. The best that can be done is to choose the nozzle angle α and the velocity of the blade with respect to the steam jet velocity so that the numerical value of V'_a will be a minimum. *For the production of maximum work the turbine blade velocity should be about one-half of the absolute velocity of the steam jet.*

227. Steam-Turbine Staging.—When the blades of a turbine are operated at peripheral speeds equal to half the steam jet velocities, one of two difficulties arises. The stresses induced in the rotor may become excessive or the rotative speed may be too high for direct connection to the machine which is to be driven. The turbine rotor may be operated at or near the theoretically desirable speed and the power transmitted to the driven machine through gears which decrease the turbine speed to the necessary extent. However, the gears necessary for the desired speed reduction or the size of the turbine wheels would be excessive in large units. In order to decrease the peripheral velocity of the turbine rotor, and at the same time use more of the energy of the steam, more than one set of rotor blades is employed. This is called compounding or staging; two methods are in use, *velocity staging* and *pressure staging*.

In *velocity staging* the steam is expanded in one set of stationary nozzles to the back pressure p_2, thus giving a relatively high velocity; the jet then passes through a succession of wheels alternating with fixed

guide passages (stators). A turbine with three velocity stages is shown diagrammatically in Fig. 189. The jet passes through the blades of the first wheel, where a part of the steam velocity is lost, as indicated by the velocity curve. Emerging from the first set of blades (rotor), the steam passes through a set of stationary blades (stator) which change the direction of flow of the steam, but not its velocity. The steam impinges on the second set of moving blades and more of the velocity is absorbed by the rotor. The direction of flow is again changed by the stator and the velocity is then reduced to a minimum by the third wheel of the rotor. The work obtainable from the steam is small after the second velocity stage is passed and a third wheel is not usually employed.

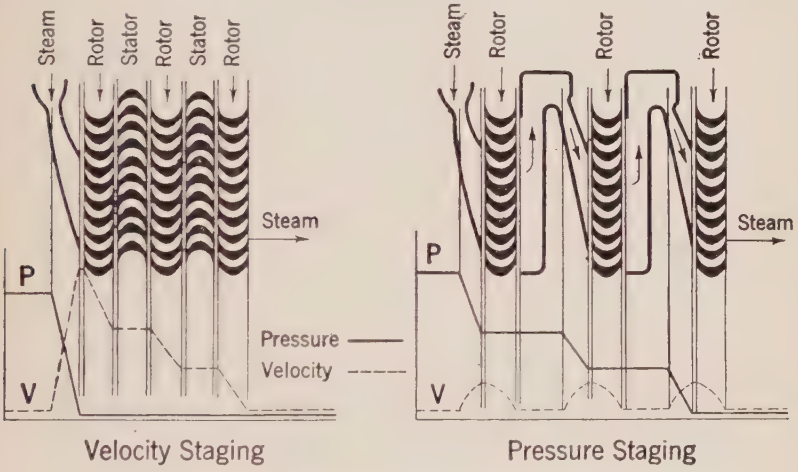


Fig. 189.—Multi-staging impulse turbines.

Steam partially expanded in a set of nozzles, and the resulting velocity utilized in moving blades, may be further expanded in a second set of nozzles with an increase of steam velocity. Moving blades adjacent to the second set of nozzles absorb the velocity as in the case of the first set. This expansion and velocity increase may be repeated several times. The process is termed *pressure staging*. Fig. 189 shows the method diagrammatically for three sets of nozzles in three pressure stages. The pressure and velocity changes of the steam are shown approximately in the graphs at the bottom of the figure. The steam, with its acquired velocity, impinges on the blades of the first wheel and the velocity is absorbed by the rotor. As the steam leaves this wheel with low velocity it passes through a second set of stationary nozzles in which the pressure is again decreased and velocity is acquired. The

second wheel absorbs this velocity and the steam passes on through the successive stages, until the pressure is reduced to a minimum.

Example.—A steam turbine of the impulse type is to be used for driving a generator at 3600 r.p.m., and the mean diameter of the rotor is to be 36 in. The quantity of dry steam to be supplied is 7200 lb. per hour at a pressure of 200 lb. per sq. in. abs. Assuming adiabatic expansion to a condenser pressure of 2 in. of Hg abs., and neglecting friction, find (*a*) the theoretical number of velocity stages required, (*b*) the theoretical number of pressure stages, and (*c*) the number of pressure stages if there are two velocity stages for each pressure stage.

Solution.—(*a*) *Velocity staging*

$$\text{Peripheral velocity} = \frac{3600}{60} \times 3\pi = 565.5 \text{ ft. per sec.}$$

$$2(1198.5 - 861.8) = 673.4 \text{ B.t.u. available per sec.}$$

$$\text{K.E.} = \frac{WV^2}{2g}$$

or

$$673.4 \times 778 = \frac{2V^2}{64.34}$$

then $$V = 4100 \text{ ft. per sec. available.}$$

Steam velocity per stage $= 2 \times 565.5 = 1131$ ft. per sec., thus

$$n_v = \frac{4100}{1131} = 3.62$$

Use 3 velocity stages.

(*b*) *Pressure staging*

Steam velocity to be absorbed per stage = 1131 ft. per sec.

$$\text{K.E.} = \frac{WV^2}{2g} = \frac{2 \times (1131)^2}{64.34}$$

$$\text{K.E.} = 39{,}800 \div 778 = 51.2 \text{ B.t.u. per stage}$$

Heat drop = 673.4 B.t.u. available, thus

$$n_p = \frac{673.4}{51.2} = 13.15$$

Use 13 pressure stages.

(*c*) *Pressure and velocity staging*

Blade velocity = 565.5 ft. per sec.

Steam velocity absorbed per pressure stage $= 2 \times (2 \times 565.5) = 2262$ ft. per sec.

Then $$\text{K.E.} = \frac{WV^2}{2g} = \frac{2 \times (2262)^2}{2 \times 32.17 \times 778}$$

$$\text{K.E.} = 205 \text{ B.t.u. per sec. per pressure stage}$$

But heat drop = 673.4 B.t.u. (available) per sec.

Therefore $\frac{673.4}{205} = 3$ pressure stages, with each pressure stage having two velocity stages.

In making a comparison of the results obtained in the preceding example, it might seem at first thought that velocity staging is the better, as a smaller number of stages is required. The advantage of the smaller number of stages is overbalanced by the fact that the pressure-stage type is more efficient. Turbines with pressure staging are used extensively in medium sizes and are often called multi-cellular because each pressure stage is composed of a compartment, that is steam tight except for the openings through the nozzles. Where no difference in pressure exists, as in velocity staging, there is no leakage. In the velocity-stage type, the pressure beyond the nozzles is the same (see Fig. 189) throughout the whole turbine casing and thus there is less interstage leakage. In a velocity-stage turbine, the efficiency is low after the first two stages, principally because the high-velocity steam jet is broken up by its passage through the blades. Therefore, more than two velocity stages per pressure stage are seldom used. A very high entering steam velocity also produces friction between the steam and the blade surfaces, causing a considerable loss.

228. Single-Pressure, Compound-Velocity-Stage Impulse Turbines. —The Elliott Form "CP," De Laval Class "C," Moore single-stage, General Electric single-stage, Terry non-condensing, and Sturtevant "Type 12" are well-known steam turbines of this class. Fig. 190 shows a longitudinal section through an Elliott "CP" single-pressure, two-velocity-stage, non-condensing turbine embodying the "Curtis" (velocity staging) principle. The rotor consists of a shaft, two wheels and the speed governor. The revolving blades and stationary guide vanes are made of nickel steel and are inserted in dovetail slots in the wheels and stationary holder. The nozzles are of the diverging type and are machined in a nozzle ring casting bolted to the side of the steam ring. The spacing is very close so that the steam jets unite upon leaving the nozzles, and the blades, therefore, receive a continuous band of steam. The intermediate stationary guide-vanes receive the steam from the blades of the first wheel and direct the steam upon the second row of blades. As complete expansion takes place in the nozzles, uniform pres-

sure exists throughout the remainder of the turbine; hence there is no tendency for leakage of steam over the tops of the blades. This fact permits the use of large running clearances. Steam leakage along the shaft is partially prevented by carbon packing rings, Fig. 190.

The speed of the Elliott " CP " turbine is controlled by a centrifugal governor mounted on the end (left end in Fig. 190) of the main shaft and attached, through suitable linkage, to a throttle valve in the steam supply line. Two governor weights are mounted upon hardened steel knife-edges in a bronze cage that rotates, the cage itself being keyed and pressed on an extension of the turbine shaft. A coil spring resting against a sliding sleeve loads the governor so that any change in the spring com-

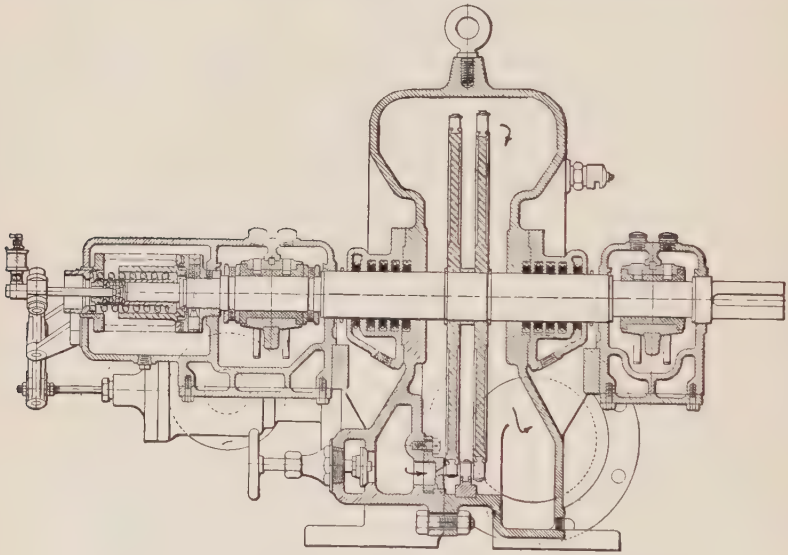


Fig. 190.—Longitudinal section, Elliott form "CP" turbine.

pression varies the speed of the turbine. The motion from the weights to the governor linkage is transmitted by a hollow spindle that does not rotate; this permits positive lubrication for the governor mechanism.

Machines suitable for high speeds, such as small high-head centrifugal pumps, centrifugal blowers and compressors, and electric generators, can be directly connected through a flexible coupling to steam turbines of this class. Reducing gears are used between the turbine and machines restricted to speeds lower than those of the turbine, as medium-size generators, large centrifugal pumps and blowers, belt pulleys, paper mill and other moderate speed machinery. Elliott Form " CP " turbines are available in sizes up to 150 b. hp.

The De Laval " Class C " turbine differs essentially from the Elliott " CP " in that the rotor consists of a single wheel fitted at the periphery with two or three rows of blades. The De Laval turbine of this class is built in sizes up to 800 b.hp. and is primarily designed for non-condensing service.

The Terry non-condensing and Sturtevant " Type 12 " turbines apply the principles of single-pressure, compound-velocity-stage turbines to a single wheel of the re-entrant (helical flow) type. The steam issues from an expanding nozzle at high velocity, enters the side of the wheel bucket (blade), where the direction of steam flow is reversed 180 deg., Fig. 191. As this single reversal uses but a portion of the available energy, the steam is received in a reversing chamber and returned to the wheel. The reversal of the steam several times to the same row of moving buckets gives the desired multi-velocity effect and the speed of rotation is reduced. The moving buckets are semicircular and are milled in the solid rim of the wheel. The stationary reversing buckets are arranged in groups around the inner surface of the turbine casing, each group being supplied with a separate nozzle, Fig. 192. The nozzles have hand-controlled valves. These permit the use of the required number for high efficiency at various loads. Each stationary bucket has a crescent-shaped hole at the center, Fig. 191, through which the steam enters the casing exhaust chamber, when the steam velocity has been sufficiently reduced. The casing is subject only to the exhaust pressure and temperature and is in reality a portion of the exhaust piping. The governor is of the throttling type, similar to the governor shown in Fig. 190.

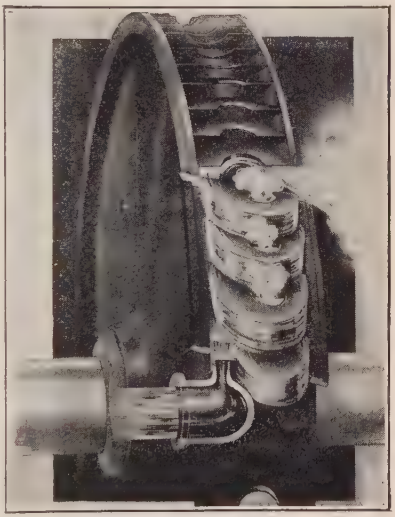


Fig. 191.—Action of steam in Terry turbine.

As Terry turbines of this type are built only in sizes from 5 to 1000 hp., their use in central stations is practically confined to driving auxiliary apparatus, such as pumps, blowers, and small generators. The non-condensing turbine is used when exhaust steam can be utilized for heating purposes or for process work in sugar plants, paper mills, tanneries, rubber mills, etc. The absence of oil in the exhaust steam makes the non-condensing turbine particularly valuable in such plants. The

turbine acts as a power-generating reducing valve and abstracts only a small percentage (about 5 per cent) of the heat in the steam.

229. Compound-Pressure, Single-Velocity-Stage Impulse Turbines.—Professor Rateau, of Paris, was the pioneer in the development of this type of steam turbine. The turbine shown in Fig. 193 is illustrative of the Rateau type of construction. The rotor consists of a number of steel disks, keyed and shrunk on the turbine shaft and held in place axially by shrink rings which fit into recesses on both sides of each wheel. The blades are rigidly fastened to the periphery of the wheel

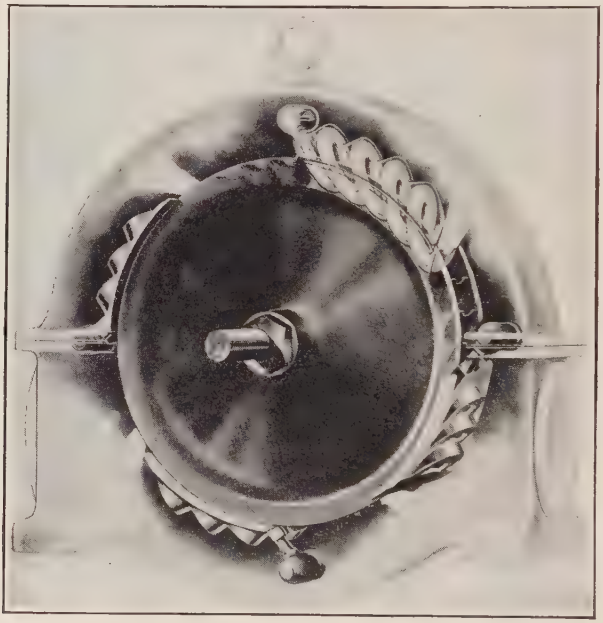


FIG. 192.—Phantom view of Terry turbine.

by bulb shanks, individually driven into slots and peened into place. The stationary element consists of the steam heads, diaphragms, nozzles, etc., as shown in Fig. 193. The cylindrical sections, end castings and diaphragms are all split on the horizontal center line so that the entire upper half of the casing may be removed for inspection without disturbing the steam and exhaust connections. Steam or air leakage along the shaft is prevented by three-part arch-bound carbon rings; held in place by non-corrosive metal springs and prevented from rotating by stops. The circular diaphragms contain the nozzles and separate the stages.

Steam admitted to the steam chest of the turbine passes through the first-stage nozzles; they occupy only a part of the circumference and may be controlled by hand-operated valves. The kinetic energy of the steam is absorbed by the blades on the first turbine wheel. After leaving the first row of moving blades at a low velocity, the steam is again partially expanded in the second set of nozzles and velocity is acquired. The second wheel absorbs this velocity and the steam passes on through the following stages, each consisting of a successive row of nozzles and blades. Ten pressure stages are shown in Fig. 193, and the pressure-velocity changes for three stages of this turbine are shown in Fig. 189. At the high-pressure end of the turbine there are only a

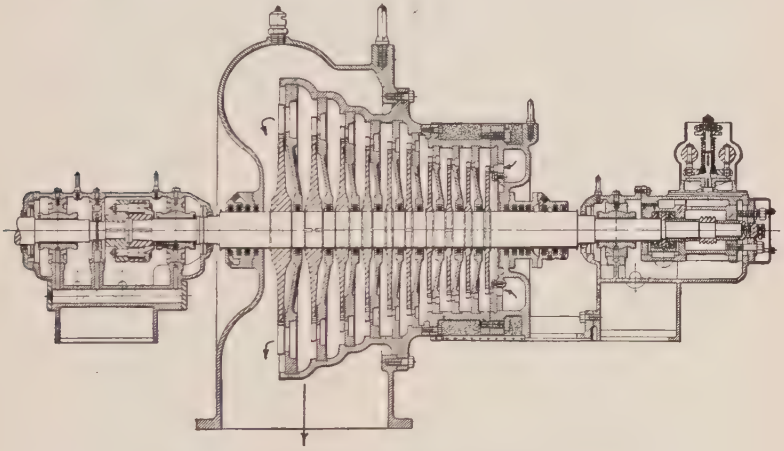


Fig. 193.—Longitudinal section, Rateau type turbine.

few groups of nozzles, but in each successive stage the number increases to allow for the increased volume of the expanded steam, until in the last stages the entire circumference of the diaphragm is used. The height of the blades increases as the exhaust end is approached, because of the increase in the volume of steam flowing.

Under ordinary conditions of operation, no appreciable end thrust exists with turbines of this type. However, the rotor must necessarily be located in a predetermined position with respect to the stationary elements. This is accomplished by using a Kingsbury thrust bearing, right end Fig. 193, made up of a number of tilting, self-aligning bronze shoes which bear against a collar keyed in a fixed position on the turbine shaft. As the collar rotates, the shoes tilt and allow a thin film of oil to be drawn in between the shoes and the collar. The load is thus "floated" on a film of oil separating the bearing surfaces. Two sets of

shoes are provided, one set on each side of the thrust collar to take thrust in either direction. The bearing housing rests upon a sliding fit, which permits endwise adjustment of the rotor while the turbine is in operation.

The governor mechanism is of the oil-operated type. The main governor, right end, Fig. 193, is of the spring-loaded centrifugal type with spherical weights and is gear driven from the turbine shaft. The governor motion is transmitted through linkage to the pilot valve of an oil-relay cylinder, which admits oil under pressure to either side of the main operating piston. The movement of this piston opens or closes a double-seated balanced governor (throttle) valve which controls the admission of steam to the turbine, and hence the speed.

230. Compound-Pressure, Compound-Velocity-Stage Impulse Turbines.—The advantages of velocity staging are: (1) a large pressure drop can be employed in each stage for a given blade velocity, (2) there is a small number of stages for a given heat drop and consequent reduction in first cost, and (3) the maximum pressure and temperature in the turbine casing are less than in pressure staging, thus decreasing stresses and interstage leakage. A high steam velocity as used in multi-stage velocity turbines is, however, a disadvantage especially if moisture is present in the steam, as rapid erosion of the blades is inevitable. In multi-stage pressure turbines the stage velocity is constant but the steam pressure is gradually decreased from inlet to exhaust. The efficiency of a purely velocity-compounded turbine is less than that of a purely pressure-compounded turbine, but the larger pressure drops permissible with the former type are an advantage. In order to utilize the low spouting velocity of the steam and higher efficiency of the pressure type, and also reduce the number of stages; both pressure and velocity staging are used in the same turbine. Ordinarily there are two velocity stages in the first pressure stage and one velocity stage for each of the remaining pressure stages.

An example of this type of turbine, as made by the Moore Steam Turbine Corporation, is shown in section in Fig. 194. The steam is expanded through the first-stage nozzles, and impinges on a Curtis (velocity-stage) wheel consisting of two rows of moving blades with a stationary row of reversing guide vanes between them. After this first-pressure stage, the steam flows successively through a series of six additional pressure stages, each consisting of a set of nozzles and a turbine wheel. As the steam passes through successive stages the passage areas increase sufficiently to provide for the increased volume of steam.

The casing of the Moore turbine is split horizontally so that the upper half can be removed to give access to the interior, and is encased

with a layer of non-conducting material covered by sheet metal. The wheels are keyed and shrunk on the shaft, which is supported by two

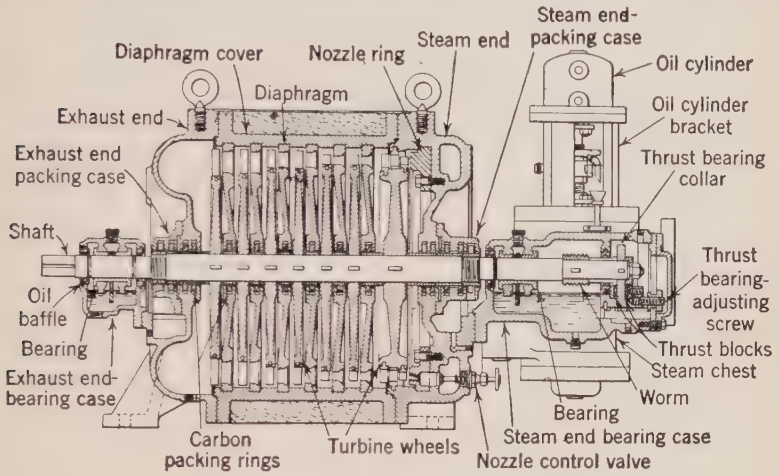


FIG. 194.—Moore multi-stage steam turbine.

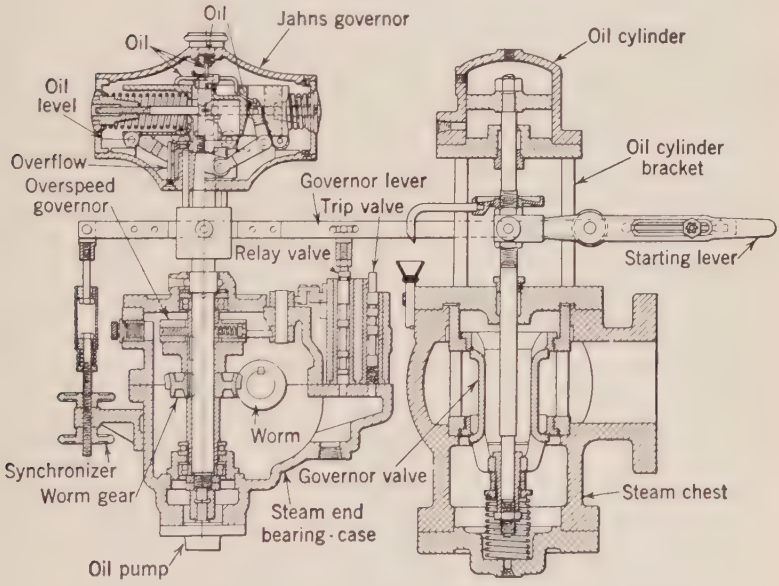


FIG. 195.—Oil-relay governor and valve gear, Moore turbine.

ring-oiled bearings. Carbon packing is used for sealing against interstage and other leakage. A Jahns governor, Fig. 195, mounted on

a vertical shaft is driven by a worm gear from the turbine shaft (right end in Fig. 194). The governor actuates the oil-relay valve which admits oil under pressure above or below the piston in the oil cylinder. This piston in turn controls the opening of the throttling governor valve. Oil under a pressure of 35 to 40 lb. per sq. in. is supplied by a geared type of pump driven from the lower end of the governor spindle. Hand nozzle-control valves are furnished: (1) for overload operation, (2) for full-load operation on steam pressures lower than the pressure for which the turbine was designed, and (3) economical operation at fractional loads.

231. Velocity Diagram for Compound-Velocity-Stage Impulse Turbine. No Losses.—The extended velocity diagram for a single-pressure, single-velocity-stage steam turbine is shown in Fig. 188. The velocity diagram for a single-pressure two-velocity-stage turbine is shown in Fig. 196. The blade velocity being the same for each turbine wheel, the series of velocity triangles have been superimposed to form one diagram. CA, the base of the velocity triangles, represents the blade velocity V_b in feet per second, while AB is the initial steam velocity, the nozzle angle being α. CB is, therefore, the relative inlet velocity and the angle β (BCM) is the inlet angle of the first row of moving blades. Since there are no losses, the relative exit velocity CD is equal to CB; and since the blades are equiangular, the angle α' is made equal to angle β. This is easily accomplished by making the axial components of the velocities equal. DA is the absolute velocity of the steam leaving the first row of moving blades, or the velocity relative to the stationary guide vanes. Hence, angle β' (DAN) is the inlet angle of the guide vanes. Assuming no losses through the guide vanes, the absolute entrance velocity EA is equal to DA and angle EAM is the outlet angle of the guide vanes. The velocity triangles for the second row of moving blades may be drawn in the same manner as for the first row. For the most efficient performance of work the final absolute vector AF should be perpendicular to the direction of motion of the blade, that is, AF and $A\ H$ should be coincident. V_2 represents the tangential change of velocity or the "velocity of whirl" of the steam in the second row of moving blades. Any number of pressure stages may be analyzed in a similar manner.

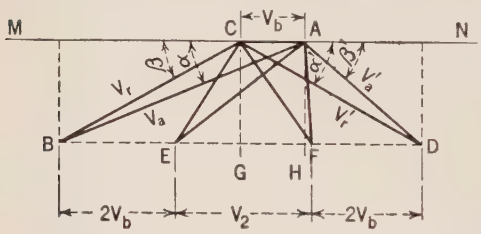


FIG. 196.—Velocity diagram for two-row velocity-stage impulse turbine.

232. Reaction Turbines.—In the pure reaction turbine the expansion of the steam occurs in the moving blades. The turbines commonly called reaction turbines use both the impulse and reaction principles. This type of reaction turbine is shown diagrammatically in Fig. 197. The steam enters from the left and passes through a set of stationary blades (stator) in which occurs some drop in pressure. In passing through the next set of blades, which are moving, a further drop in pressure occurs. The velocity acquired in the stationary blades and also that produced in the rotor blades is absorbed by the rotor blades. The jet with low residual velocity is deflected to the second set of stationary blades with another drop in pressure (the reduction seldom exceeds 3 lb.), and then enters the second rotor at practically the same velocity as that of the blades. The pressure and velocity curves in Fig. 197 show the changes occurring in turbines of this type as the steam passes alternately through fixed and moving blades. Considerable expansion takes place in the moving blades and consequently, because the pressure is not the same on both sides of these blades, the turbine is called a reaction turbine.

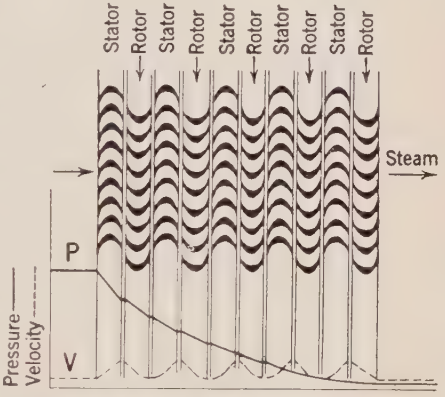


FIG. 197.—Multi-stage reaction turbine.

Example.—A steam turbine of the reaction type is to be used for driving a generator at 3600 r.p.m.; the mean diameter of the rotor is to be 36 in. Assuming adiabatic expansion and neglecting friction, find the number of stages required in a "pure" reaction turbine and in an "impure" reaction turbine if 7200 lb. of dry steam are supplied per hour at a pressure of 200 lb. per sq. in. abs., the condenser pressure to be 2 in. mercury abs.

Solution.—In order to obtain the highest efficiency, the velocity of the moving blades of a reaction turbine should be approximately the same as that of the steam relative to the blades.

$$V_b = \frac{3600}{60} \times 3\pi = 565.5 \text{ ft. per sec.}$$

and the steam velocity = 565.5 ft. per sec.

$$\text{K.E.} = \frac{WV^2}{2g} = \frac{2 \times (565.5)^2}{2 \times 32.17} \times \frac{1}{778}$$

$$\text{K.E.} = 12.8 \text{ B.t.u. heat drop per stage per sec.}$$

Total heat drop = (1198.5 − 861.8)2 = 673.4 B.t.u. per sec.

$$\textit{Pure} \text{ reaction stages} = \frac{673.4}{12.8} = 52$$

For the pressure-stage impulse type (with the same data) in the illustrative problem of Art. 227, the number of stages required was 13. Since the "impure" or actual reaction turbine employs a combination of the pressure-stage impulse and the reaction principle, the number of stages may be taken as a mean or

$$\textit{Impure} \text{ reaction stages} = \frac{52 + 13}{2} = 32$$

The preceding example and solution show that the reaction turbine requires many more stages than the impulse type. This is due to the fact that a much higher ratio of blade velocity to steam velocity is required, for the best efficiency of a reaction stage, than is required for an impulse stage. However, the loss due to friction is less in the reaction type since the steam velocities are comparatively low; also erosion of the blading is reduced to a minimum. The axial-flow impulse reaction turbine, Fig. 197, is called the Parsons turbine, after its inventor, Sir Charles A. Parsons. The Allis-Chalmers Manufacturing Company, the Westinghouse Electric and Manufacturing Company, and the American Brown Boveri Corporation are the important American manufacturers of this type of turbine. In the radial-flow reaction turbine, the steam flows outward from the shaft to the periphery of the rotor, in this way accommodating the increased volume due to expansion. The Ljungström (Swedish) is the best known of this type of reaction turbine.

The Allis-Chalmers high-pressure condensing turbine is a single-cylinder machine using the reaction principle (axial flow) throughout all stages. Fig. 198 shows a longitudinal section through an elementary Allis-Chalmers steam turbine. The turbine consists essentially of a fixed casing or cylinder and a revolving spindle or drum. The ends of the spindle are extended in the form of a shaft carried in two bearings, A and B. Steam enters from pipe C after passing through the main regulating valve, which is under the control of the governor. Steam enters the cylinder through passage E, turns to the left, passes through alternate stationary and moving blades, and finally is exhausted to the condenser through connection G.

Each row of blades, both stationary and revolving, extends completely around the turbine and the steam flows through the full annulus between the spindle and the cylinder. In an ideal turbine the lengths of the blades and the diameter of the spindle which carries them would continuously and gradually increase from the steam inlet to the exhaust.

Practically, however, the desired effect is produced by making the spindle in steps, there generally being three such steps or stages, H, J, and K. The blades in each step are arranged in groups of increasing length. At the beginning of each of the larger steps the blades are usually shorter than at the end of the preceding smaller step, the change being made so that the correct relation of blade length to spindle diameter is secured.

The difference in steam pressure at the end of each step produces a thrust tending to force the spindle toward the left, Fig. 198. This thrust is counteracted by the balance pistons L, M, and N, which are of the necessary diameter to neutralize the thrust on the spindle steps

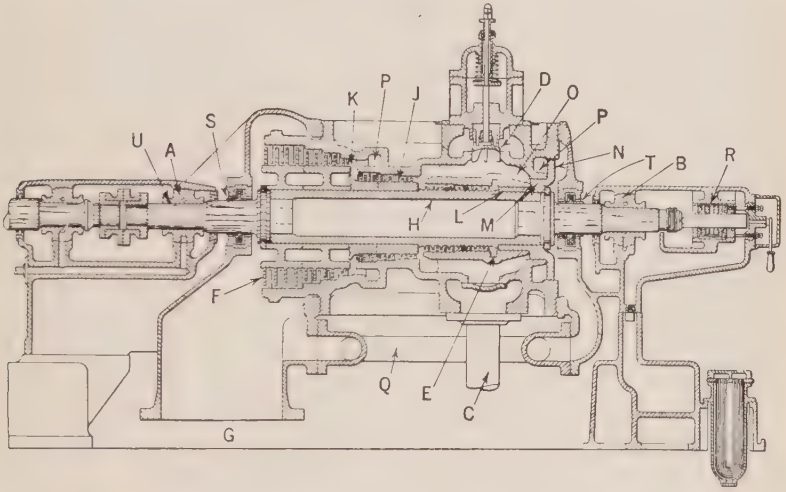


Fig. 198.—Section through an elementary Allis-Chalmers steam turbine.

H, J, and K, respectively. The elements are called pistons, although they do not come in contact with the cylinder, but both the pistons and the cylinder are provided with alternate rings which form a labyrinth packing to retard the leakage of steam. In order that each balance piston may have the proper pressure on both sides, equalizing passages O, P, and Q are provided, connecting the balance pistons with the corresponding stages of the blading. The end thrust being thus practically neutralized by means of the balance pistons, the spindle "floats" so that it can easily be moved axially. In order to fix the position of the spindle definitely, a small adjustable collar bearing is provided at R. This collar bearing is used to locate and hold the spindle in a position to give the proper clearance between the rings of the balance piston and those of the cylinder. Leakage of steam is reduced to a minimum, and

at the same time actual contact between moving and stationary parts is prevented under varying conditions of temperature.

Where the shaft passes through the cylinder casing, at S and T, it is necessary to provide against in-leakage of air, or out-leakage of steam in the case of temporary operation non-condensing. In the Allis-Chalmers turbines these glands are practically frictionless, being made tight by water packing without metallic contact. The turbine shaft is extended at U and coupled to the shaft of the alternator by means of a flexible connection. The valve D is provided for admitting steam to the intermediate stage of the turbine and is arranged to open when the main inlet blading cannot pass a sufficient quantity of steam. The purpose of valve D is to allow the turbine to carry momentary overloads, also to keep the machine at speed and carry its load should the steam pressure drop below normal. This valve is operated by the turbine governor. The oil pump, operated from the turbine shaft by gears working in an oil bath, is located at the thrust end of the turbine below the level of the oil tank. The pump delivers oil at a pressure of from 25 to 30 lb. per sq. in. to the oil cooler (lower right in Fig. 198), where the temperature of the oil is reduced before passing on to the bearings, etc. After leaving the cooler, part of the oil is used for the oil-relay system of governing the turbine and the remainder is delivered to the various bearings at a reduced pressure of about 3 lb. per sq. in.

The *velocity diagram* for the first stage of a reaction turbine is shown in Fig. 199. In a reaction stage there is an increase of relative velocity

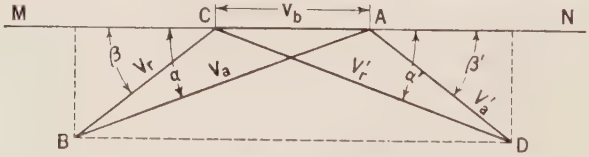


Fig. 199.—Velocity diagram for reaction turbine blades.

between entrance and exit, due to the pressure drop in the blade channels. The fixed and moving rows have identical blades and the heat drop for the stage is divided equally between the stationary and moving rows of blades. Assuming the same friction losses in identical rows, it follows that with the same exit angle for each, $\alpha = \alpha'$, the entrance and exit triangles are identical, as shown in Fig. 199. Thus $V_r = V'_a$ and $V_a = V'_r$. From the velocity diagram it is evident that the axial components of the steam velocity are equal and the dynamical thrust is zero. With the usual construction, however, in which the blade heights are kept constant throughout each step or group, there is a progressive increase of velocity and consequently a slight thrust. Normally, this

thrust is small and is neglected. There is a statical thrust, which does not occur in the impulse turbine, due to the difference of pressure between entrance and exit, at each rotor. The cumulative effect of these statical thrusts is counteracted by balance pistons, as shown at L, M, and N, Fig. 198.

233. Combination Steam Turbines.—The high-pressure portion of the reaction turbine is comparatively inefficient because of blade tip leakage, while the low-pressure part is usually efficient. The impulse turbine has approximately the same efficiency at all loads, if governing by nozzle control is used. The replacement of the high-pressure section of the reaction turbine by impulse staging very materially shortens the rotor, thus increasing its stiffness. Distortion of the turbine casing and rotor is avoided by such an arrangement, as the entering steam has a considerable reduction of temperature and pressure in the nozzles before reaching the rotor. A *combination turbine* is one in which impulse blading is used for the high-pressure steam and reaction blading in the low-pressure stages.

Fig. 200 shows a section through a combination turbine, which is typical of some recent installations of large single-cylinder machines. Steam passes first through five single-row impulse wheels separated by diaphragms in the usual manner, and then through five reaction wheels, each carrying one row of blades. In the smaller combination turbines, 1000 to 10,000 kw., this manufacturer uses a high-pressure impulse wheel of comparatively large diameter, with one or more velocity stages and partial peripheral admission. The reaction section which follows has full admission and the diameter of the drum is comparatively small. The drum type rotor, similar to the rotor of Fig. 198, is employed in these (smaller) turbines. A solid type rotor, such as is used in turbines for small outputs, would not be suitable for large turbines as the stresses would be high due to the increased diameter of the rotor. To prevent excessive rotor stresses and to realize the efficiency of the conical-flow type turbine, the reaction blading is placed on disks mounted on the turbine shaft, Fig. 200. In the larger turbines the volume of steam in the reaction stages is very great so that the diameter at the low-pressure end of the turbine must be enlarged in order to insure sufficient sectional area for the steam passage. The dimensions of the large combination turbines are comparatively small, particularly the rotor, the effective length of which is 42 in. for a 15,000-kw. unit running at 3600 r.p.m. The short length is mainly attributable to the absence of diaphragms in the reaction staging, so that the distance between the individual disks is made only large enough to be consistent with strength.

Large turbines require special precautions to prevent unequal

expansion of the casing. The Brown Boveri turbine, Fig. 200, has a number of ribs provided and arranged to counteract any tendency to distortion due to unequal temperature at the inlet and exhaust ends of the

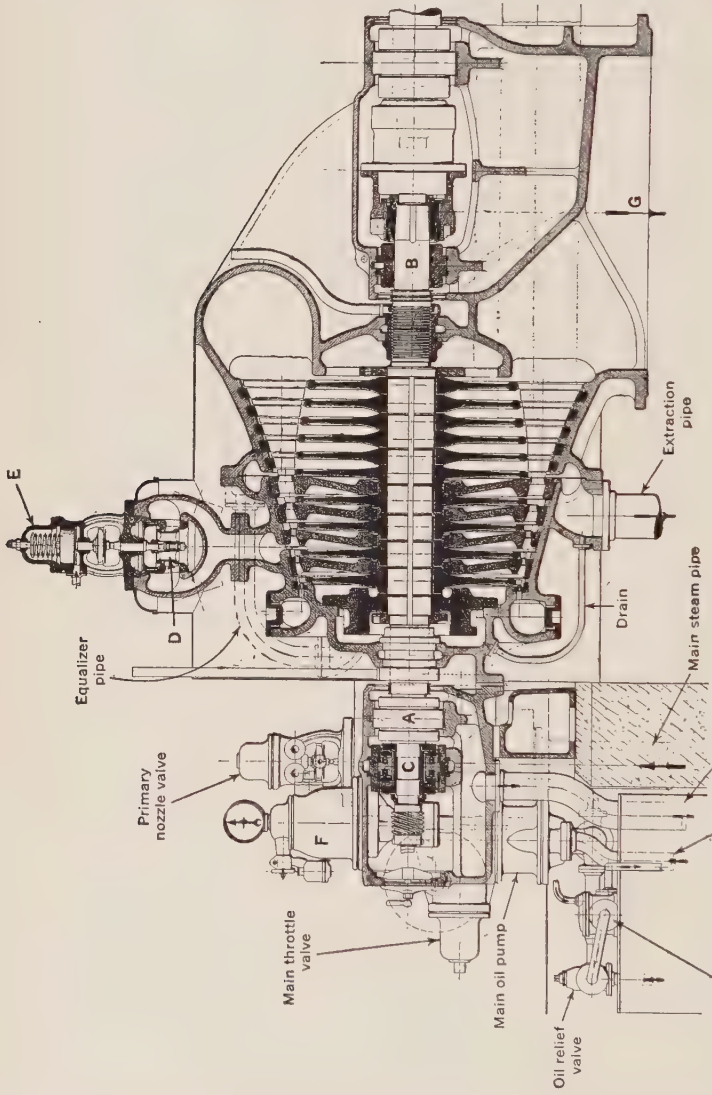


FIG. 200.—Longitudinal section, 20,000-Kw. Brown Boveri combination steam turbine.

casing. The main bearings A and B (babbitt lined) are supplied with oil under a pressure of 40 lb. per sq. in. from an oil pump bolted to the bottom of the turbine casing and driven by a worm gear on the left end of the turbine shaft, Fig. 200. A segmental thrust bearing C, to

take up any uncompensated axial thrust, is also carried by the bearing pedestal at the high-pressure end of the casing. Only one balance piston, placed left of the first wheel, Fig. 200, is necessary on account of the location of impulse stages in the high-pressure part of the turbine. Labyrinth shaft packing glands are provided at the admission and exhaust ends of the turbine. Both glands are normally under vacuum and are continuously supplied with a small amount of live steam in order to prevent air from entering the turbine. The blades of the impulse part of the turbine are made of nickel steel and, being subjected to superheated steam, the blades corrode very little. The moving blades of the low-pressure reaction stages are made of stainless steel (14 per cent chromium) and the stationary reaction blades are made of brass.

The turbine, Fig. 200, is equipped with two main automatic throttle valves. These valves are so arranged that they are tripped (closed) simultaneously by an automatic tripping device, in case of overspeed. Steam is supplied through automatically operated double-beat primary nozzle valves (usually three), actuated by the piston in an oil-relay cylinder, similar to *E*. The steam next passes through the nozzles to the impulse blading and is then expanded in the reaction sections to the condenser pressure and exhausted to the condenser through *G*. Overload valve *D* is provided for admitting steam between the second and third impulse stages. Valve *D* opens when the primary nozzle-valves cannot pass a sufficient quantity of steam to carry momentary overloads, to maintain a constant speed, and to enable the turbine to carry the load should the steam pressure drop below normal. The turbine is equipped with an extraction pipe for the removal of some of the expanded steam at about atmospheric pressure and the extracted steam is used for heating or other industrial purposes (Art. 236). The enclosed centrifugal governor *F* controls the two main throttle valves, the primary nozzle valves, and the overload valve *D*. Governor *F* is driven from the main turbine shaft by worm gearing. Oil under pressure from the lubricating oil-pump fills the pipe leading to the space beneath the piston at *E*. The pipe on the supply side is connected to a chamber formed by an adjustable sleeve on the governor spindle. The sleeve can cover and uncover an oil bypass port, according to the speed fluctuation, and the governor sleeve can be raised or lowered in consequence. A quantity of oil escapes and the oil pressure under the piston *E* is changed accordingly. When the sleeve on the governor spindle reduces the escape area of the bypass port the oil pressure under the piston increases. The increased pressure forces up one or more of the relay pistons, according to the predetermined setting of the control springs, opens the nozzle valves and overload valve successively and admits the necessary steam to meet

the increased load on the turbine. One notable feature about the relay governing system on this turbine is the absence of lever connections between the governor F and the oil-relay cylinders of the valves. The equalizer pipe allows the equalization of the steam pressure on the outboard side of the dummy piston with the pressure in the intermediate portion of the turbine. Any steam leakage past the dummy piston flows through the equalizer pipe to the intermediate portion for further utilization in the later stages of the turbine.

234. Multi-cylinder Steam Turbines.—*Multi-cylinder* or *compound turbine* is the term applied to turbines in which the steam passes in sequence through one or more separate units (cylinders), partially expanding in each. The steam passes first through the high-pressure cylinder and then is led to the low-pressure cylinder. If the high- and low-pressure units are placed on the same shaft, the machine is called a *tandem-compound turbine*. Sometimes better results are obtained by mounting each unit on its own shaft and running the two at different speeds. With the latter arrangement, the machine is called a *cross-compound turbine*. In the compound design the temperature range is less in each cylinder than in a single-cylinder turbine operating between the same initial and final steam conditions, and the stresses in the casing and rotor are decreased accordingly. Three units in series may also be used, the high-pressure and intermediate-pressure cylinders usually being in tandem and the low-pressure unit in a separate cylinder driving a separate generator. Multi-cylinder turbines are also being built with three units in tandem.

Dry steam is desired throughout the range of expansion in any turbine. The use of multi-cylinder machines may make necessary *reheating* of the steam to increase its temperature after passage through a part of the turbine. A 50,000-kw. Westinghouse turbine, at the Crawford Avenue (Chicago) Station has the high- and intermediate-pressure cylinders arranged as a tandem-compound unit, driving a generator of 30,000-kw. capacity and a 2000-kw. house generator. The low-pressure turbine, which is placed alongside the tandem element, is direct-connected to a 20,000-kw. generator. The exciter for the entire machine is also driven by the low-pressure element. Both units operate at 1800 r.p.m. At full load the high-pressure cylinder expands the steam from 550-lb. gage and 725 deg. fahr. to 132 lb. abs. with practically no superheat. The entire volume of steam is conducted to a reheat boiler and is returned to the intermediate-pressure cylinder at 122 lb. abs. and 700 deg. fahr. The steam is expanded to 16.5 lb. abs. and then passes directly to the low-pressure cylinder, from which it is exhausted at 1 in. mercury abs. Steam is extracted for feedwater heating from the intermediate cylinder

at 50 and 17 lb. abs. and from the low-pressure cylinder at 4 lb. abs. The arrangement of the high- and intermediate-pressure turbines is such that the steam flow is opposed so that one may neutralize the thrust of the other, eliminating the necessity for large balance pistons. The low-pressure element is of the double-flow type, the steam entering at the center of the rotor and flowing both ways into a 65,000-sq. ft. vertical condenser. Reaction blading is used throughout, and the last two rows of blades have a divided exhaust arrangement, a scheme for reducing the length of the low-pressure blades, originally developed by K. Baumann of Manchester, England.

Fig. 201 illustrates the triple-compound turbine-generator installed in the Philo Station of the Ohio Power Company. The high-pressure

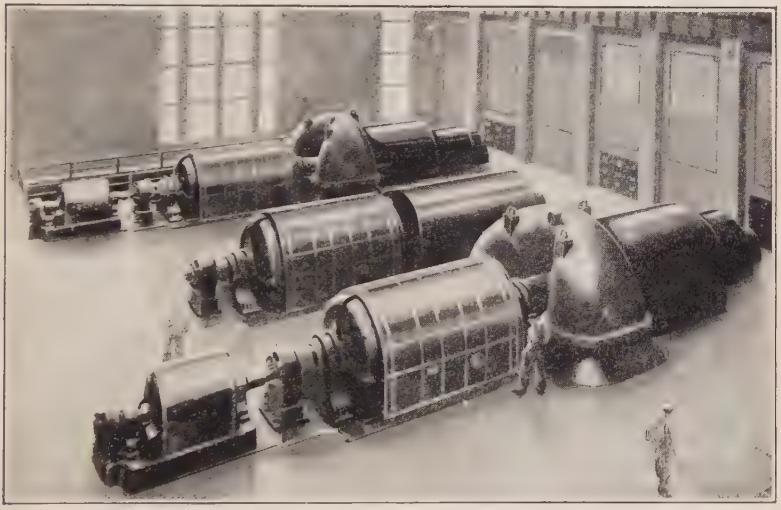


Fig. 201.—165,000-Kw. triple-compound General Electric turbine, Philo (Ohio) Station

cylinder (center) receives steam at a pressure of 600-lb. gage and a temperature of 725 deg. fahr. The exhaust steam is reheated at a pressure of 110-lb. gage to the original temperature; one-half of this reheated steam is expanded in each of the two low-pressure cylinders. Each generator has a capacity of 53,000 kw. at 1800 r.p.m. and each low-pressure turbine drives a 3000-kw. service generator. The 208,000-kw. turbine, Fig. 202, of the State Line Generating Station in Indiana is also a three-unit set operating at 1800 r.p.m. with one high-pressure (76,000 kw.) and two low-pressure cylinders (66,000 kw. each). The turbine is designed to operate with steam at 600-lb. gage, 750 deg. fahr. at the throttle, the steam being reheated between cylinders at a pressure of 95-lb. gage to 500 deg. fahr. by live steam. The low-pressure cylinder,

Fig. 203, is of the double-flow type. Steam enters at I, divides and flows in two directions, thus practically eliminating end thrust and reducing the size of the rotor blades. The steam on leaving the last stage wheels flows practically horizontally to the right and left into exhaust hoods. The steam is again divided, Fig. 202, in the exhaust hoods; thus each low-pressure turbine exhausts into four condensers (vertical) having a surface of 22,000 sq. ft. each. The stream-line exhaust hoods are designed to give a higher vacuum at the turbine wheels than at the condenser inlet, due to the diffuser action.

235. Back-Pressure Turbines.—The name of back-pressure turbine is generally given to high-pressure turbines that operate non-condensing. The steam pressure required for heating or industrial purposes may

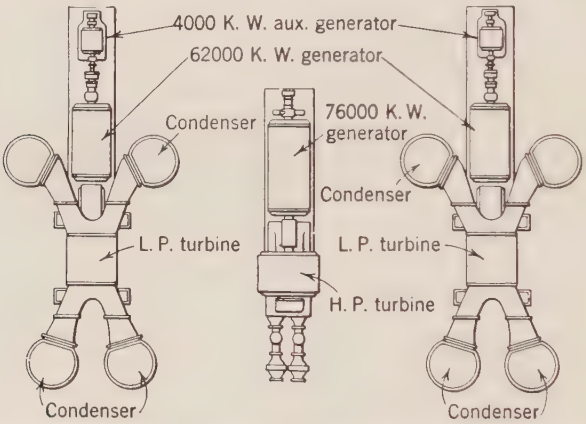


Fig. 202.—Arrangement of 208,000-Kw. turbine-generator unit, State Line Power Station.

range from atmospheric to 100 lb. per sq. in. gage pressure or more. However, the steam consumption of the turbine for a given load increases with the increase of back pressure.

Some industrial plants obtain their low-pressure steam by passing high-pressure steam through reducing valves. Back-pressure turbines serve as rotary pressure-reducing valves whereby power is delivered as a by-product in the production of low-pressure steam. The low-pressure steam is delivered to the exhaust, clean and suitable for any kind of heating. As only a small portion of the available heat is converted into power by the turbine, the remaining heat in the steam is still available for process heating or the heating of buildings. Paper mills, oil refineries, soap factories, rubber mills, tanneries, textile mills, sugar refineries, chemical plants, canneries, laundries, etc., make use of low-pressure steam in some phase of their processes.

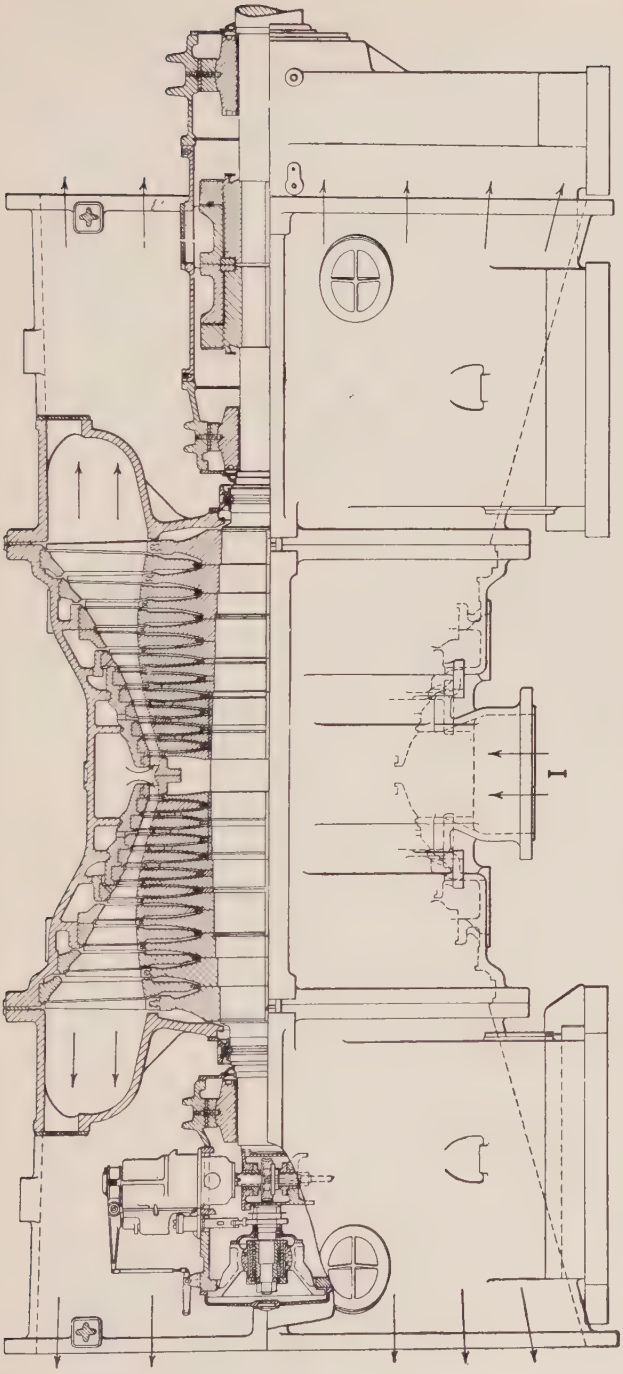


FIG. 203.—Low-pressure double-flow element of State Line 208,000-Kw. turbine.

The problem of governing is particularly simple, if the turbine is operating in parallel with other turbines which take care of governing the station output. The steam inlet valve is then controlled by a pressure governor which regulates to give a constant back pressure and allows as much steam to pass through the turbine as is needed for heating. A speed governor is also provided, and functions when the total output of the station falls below the maximum output of the back-pressure turbine. The addition of an extra valve for sending live steam into the exhaust main is desirable. Such an arrangement acts independently of the opening of the turbine steam-inlet valve, when the exhaust steam pressure has attained a certain minimum value. There are installations in which the demand for heating steam is occasionally larger than the quantity of steam passing through the turbine at maximum output. On the other hand, when no heating steam is required while power is needed, the turbine should be capable of operating with atmospheric exhaust.

A 1000-kw. turbine operating at 185 lb. per sq. in. gage initial steam pressure (dry saturated) and 8 lb. per sq. in. gage back pressure, the exhaust steam being used for heating purposes, showed steam consumptions as follows:

Full load, 37.8 lb. per kilowatt hour.
Three-fourths load, 40.4 lb. per kilowatt hour.
Half-load, 42.5 lb. per kilowatt hour.

236. Bleeder Turbines.—The field of application of the back-pressure turbine is limited, as in some industries the demand for low-pressure steam is not constant, that is, the demand for heating may have no direct relation to the demand for electrical energy. In such cases, in order to obtain greater flexibility and better efficiency of the combined power and heating plant a *bleeder* or *extraction turbine* is used. A bleeder turbine is a high-pressure condensing turbine in which provision is made for the extraction of the amount of steam required for heating or other industrial purposes, from an intermediate stage of the turbine. Partially expanded steam in any quantity from zero to 100 per cent of the entering steam may be removed from a turbine even though the demand for bled steam is variable. Suitable automatic or hand-operated valves are used to control the quantity of steam so extracted.

Fig. 204 shows a 3000-kw. Westinghouse bleeder turbine with a diaphragm or partition between the high-pressure impulse and the low-pressure reaction stages of the turbine. Ports through this partition are opened and closed by the governor-controlled butterfly valve. The

movements of the butterfly valve and the diaphragm are so correlated that as the ports of the partition are opening the butterfly valve is

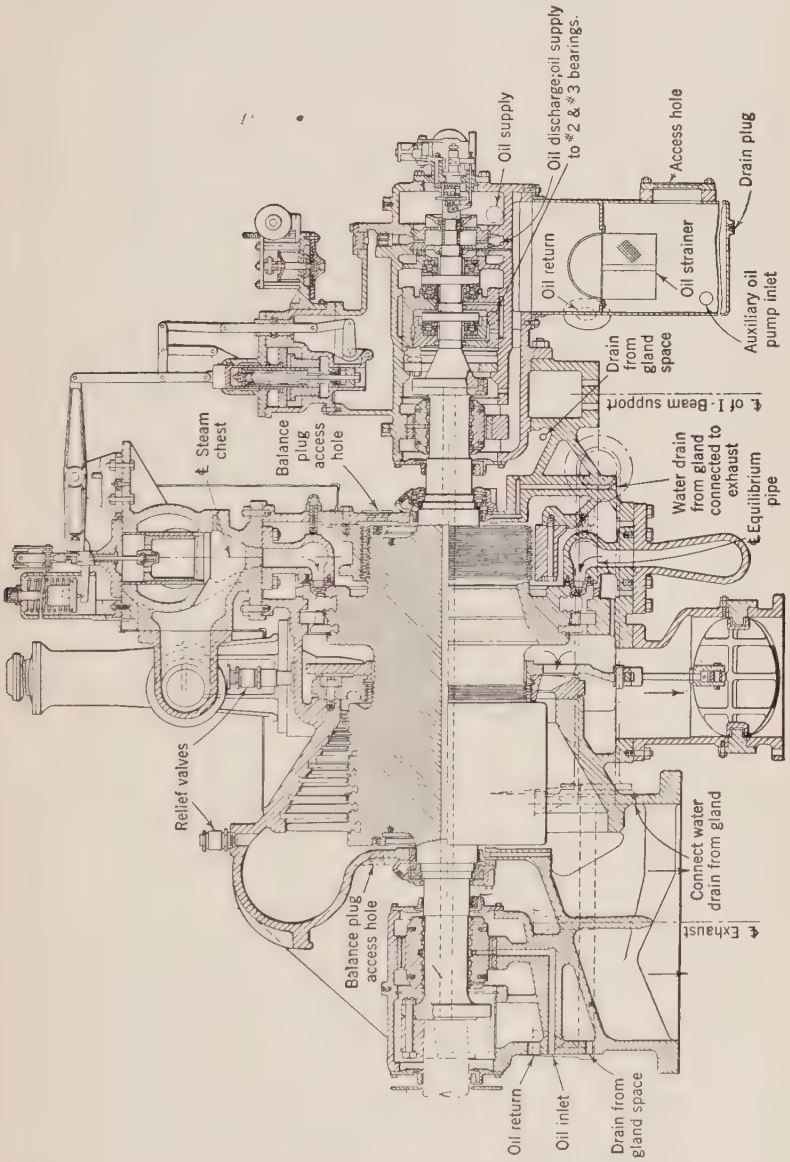


FIG. 204.—Westinghouse 3000-Kw. combination turbine, bleeder type.

closing. When extracted steam is not required, the governor closes the butterfly valve and opens the partition ports; the turbine then operates as a high-pressure condensing unit. When the ports are partially closed,

the pressure in front of the diaphragm rises and more steam is discharged into the bleeder pipe connection through the partially opened butterfly valve. When the partition ports are closed and the turbine is operating straight non-condensing, the low-pressure stages are revolving in a partial vacuum, thus relieving the turbine of the drag which the inactive stages would otherwise cause.

Provision has been made on all recent large turbines for the extraction of steam at various stages for use in feedwater evaporators, heaters and deaerators. This method, for the more complete utilization of the heat of the steam, is known as the *regenerative cycle*. Turbines have been bled at as many as four points and such procedure results in substantial reductions in the heat consumption of the plant as a whole. At the Richmond Station of the Philadelphia Electric Company a 50,000-kw. tandem-compound impulse turbine with a total of 20 stages receives steam at 390 lb. per sq. in abs. and a temperature of 675 deg. fahr. Steam from the tenth stage at 130 lb. abs. is taken to a feedwater evaporator, steam from the twelfth stage at 79 lb. abs. goes to feedwater heaters and evaporators. Steam from the fifteenth stage at 28 lb. abs. goes to feedwater heaters and a deaerator, and from the eighteenth stage at 5.6 lb. abs. to a feedwater heater.

The curves in Fig. 205 show the influence of the amount of steam extracted on the hourly steam consumption of a 1000-kw. bleeder turbine. The unit water-rate curve for condensing operation is also shown. The unit water rate at any load, for any amount of extraction, and the heat content of the bled steam may be readily calculated from the diagram.

237. Low-Pressure Steam Turbines.—The reciprocating steam engine utilizes economically a greater amount of the available energy of the steam at high pressures than at low pressures. The turbine, on the other hand, has the advantage at low pressures and in cases where large ratios of expansion are used. There is about the same amount of energy available for doing work in expanding from a medium boiler pressure down to that of the atmosphere, as there is in expanding from atmospheric pressure to that of a good vacuum. However, the amount of expansion is much larger in the second case. To utilize this expansion at low pressure the steam-engine cylinder would be very large and a large cylinder loss would occur. With the turbine the reverse is true, that is, the turbine suffers relatively less loss at low pressures than at high pressures.

The ability of the turbine to handle low-pressure steam to advantage has given rise to the use of *low-pressure turbines* in many different ways. As examples, the exhaust of hoisting engines, steam hammers, and

other apparatus, commonly exhausting at or near atmospheric pressure, may be led to one or more low-pressure turbines. Further expansion in the turbines permits the recovery of a large amount of power from what would otherwise be waste steam. On account of the intermittent operation of hoisting engines, steam hammers and rolling mill (steel plant) engines, the low-pressure turbines do not receive steam continuously. This difficulty may be overcome by using a *steam accumulator*. Such a device consists of a closed vessel interposed between the hoisting engine and the low-pressure turbine. The accumulator contains suffi-

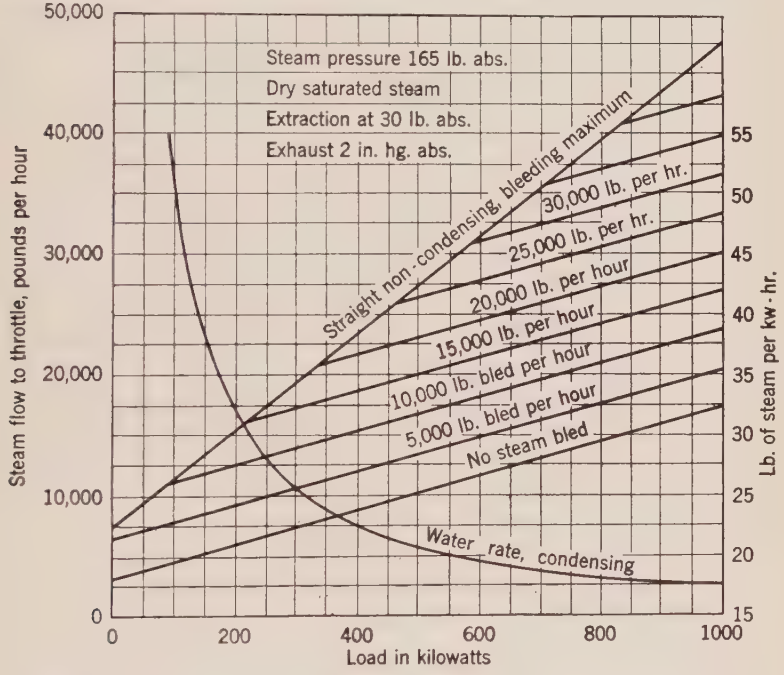


Fig. 205.—Typical curves showing steam consumption of 1000-Kw. bleeder turbine.

cient water to condense the entering excess quantity of exhaust steam. During the lowered pressure existent at the intervals of reduced steam supply, vaporization of the stored water at the expense of the contained heat augments the steam supply to the turbines.

238. Mixed-Pressure Turbines.—A mixed-pressure turbine may be regarded as a low-pressure turbine with a number of high-pressure stages. The latter enable additional high-pressure steam to be used, when the maximum available amount of low-pressure steam is insufficient to take care of the power requirements. A common type of mixed-pressure turbine consists of an impulse section in which the heat drop from boiler

to about atmospheric pressure is absorbed, and the remaining available energy is removed by expansion in low-pressure reaction blading. The valves under the control of the governor are adjusted so that no high-pressure steam is admitted until the valves on the low-pressure steam line are wide open. In some mixed-pressure turbines the high-pressure section, if of a simple disk construction, is frequently made removable so that the "windage" loss (about 1 to 3 per cent), due to rotation when not in use, can be eliminated. Mixed-pressure turbines are being built in sizes up to 3000 kw. with a steam consumption of from 20 to 35 lb. per kw. per hr., depending upon the size, initial steam conditions, per cent of rating developed, and the turbine back pressure.

239. Losses in Steam Turbines.—The most important losses that occur in steam turbine operation are: steam leakage, wheel and blade rotation, residual velocity of the steam leaving the blades, heat content of exhaust steam, radiation and friction.

To prevent leakage of steam out of or air into the turbine, where the shaft passes through the ends of the casing, either a ring-packed or water-sealed gland or a steam-packed labyrinth is used. In impulse turbines there is steam leakage between the diaphragms and the shaft, in reaction turbines leakage occurs over the ends of the blades through the radial clearance passages. This leakage loss averages about 10 per cent in most turbines. There is: a frictional resistance between the steam and the sides of a nozzle, a loss due to impact as the steam enters the moving blades, eddying and fluid friction in the blades. The windage loss due to the friction of the turbine wheel and blades revolving in steam may in some cases be very great. Tests have shown that this loss depends upon the density and quality of the steam and the speed of the wheel. If dry saturated steam is used it will be wet after expansion and the presence of the moisture will greatly increase the friction. The greater economy which results from the use of superheated steam is doubtless due to the reduced frictional resistance resulting from the absence of water in the steam. Radiation and conduction losses are usually very small because of effective covering of the turbine casing and piping with heat-insulating material. Other losses include bearing friction, friction of the gears (if any are used), and power for the governor and oil pump.

240. Steam Turbine Performance.—The history of the development of large turbo-generators is shown in Fig. 206. It is interesting to note that as the capacity and steam pressure have been increasing the heat used per kilowatt-hour has decreased until the thermal efficiency of the modern large unit is about 30 per cent. The steam turbine is well adapted to the driving of electric generators of the alternating-current

type as the desirable speeds of the two machines are the same. Such units can be constructed in large capacities with a high thermal efficiency at a comparatively low cost per kilowatt. These advantages have resulted in bringing the steam turbine into almost universal use where steam is used for producing power with units in excess of 3000 kw.

Comparisons of the steam consumption of turbines and reciprocating engines in a general way are difficult because of the diversity of operating conditions. However, the steam consumption of the better varieties of the two types is of the same order for comparable conditions with the

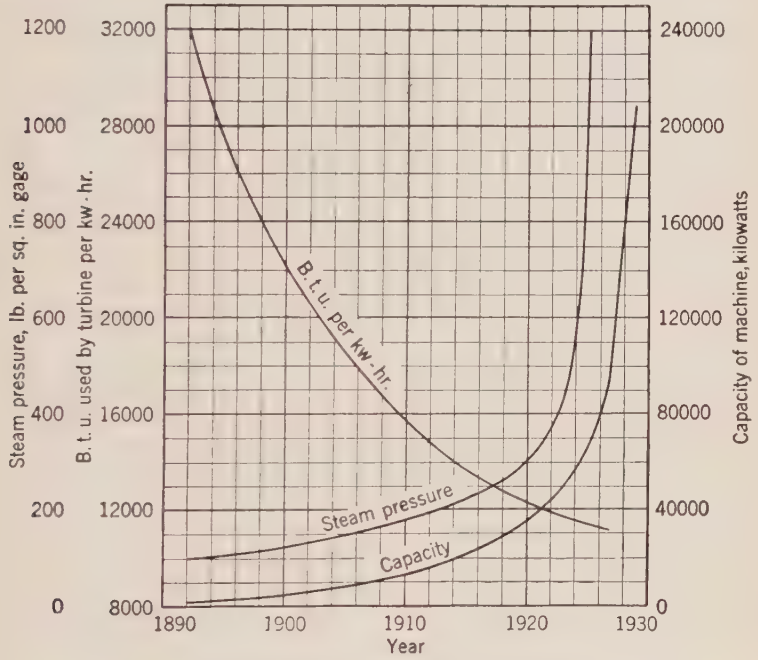


Fig. 206.—Trend of steam turbine progress.

advantage probably slightly in favor of reciprocating engines in the smaller sizes and for non-condensing service but in favor of the turbine in the larger sizes for condensing service. The steam turbine operates on the complete expansion cycle while the reciprocating engine operates on a cycle with incomplete expansion. The turbine therefore has a certain theoretical advantage because its cycle is such as to convert into work a greater amount of heat per pound of steam used between given upper and lower pressures.

A comparison of the curves in Fig. 207 shows the advantage of using manually controlled nozzle-valves for economic operation of a turbine

at partial and full loads. For an output of 30 kw. two nozzles should pass sufficient steam, whereas for 60 kw., only four nozzles should be

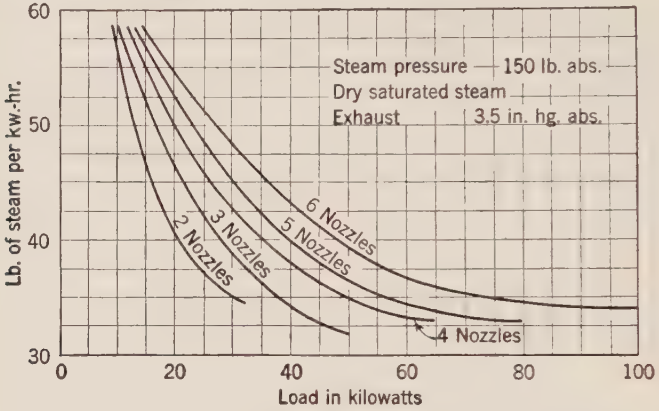


Fig. 207.—Steam consumption curves for 100-Kw. steam turbine.

open for best economy. Fig. 208 shows the results of a test on a 30,000-kw. turbine, with least steam consumption and the highest thermal efficiency at 80 per cent of the rated load. However, the econ-

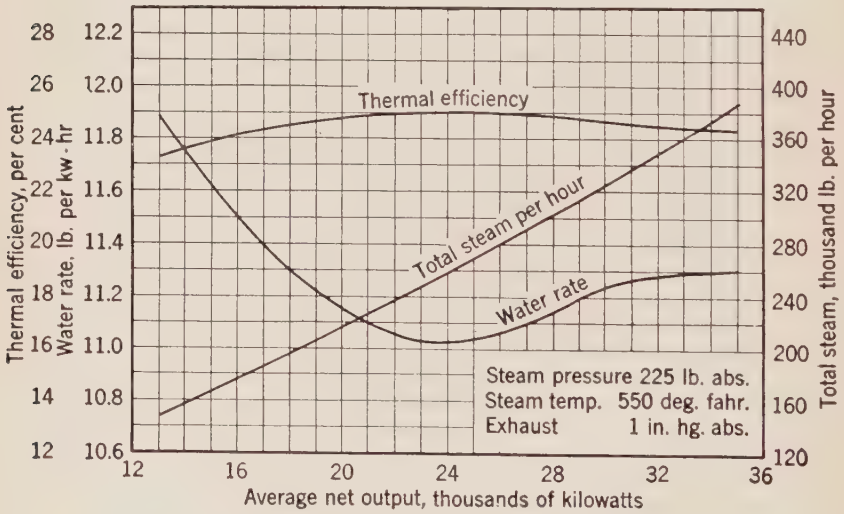


Fig. 208.—Performance curves of a 30,000-Kw. steam turbine.

omy and efficiency curves of this turbine do not vary greatly between 18,000- and 35,000-kw. output, thus insuring good economy over a wide range of load.

PROBLEMS

1. Steam expands adiabatically in a frictionless nozzle from an initial pressure of 250 lb. per sq. in. abs., steam temperature 450 deg. fahr., to a back pressure of 18 lb. per sq. in. abs. Weight of steam discharged 1800 lb. per hr. (*a*) Find the theoretical velocity of the jet of steam at the throat and exit of the nozzle. (*b*) What is the quality of the steam at throat and exit? (*c*) Determine the diameter of the nozzle at throat and exit, also the length of the nozzle from throat to exit.

2. A turbo-generator delivers 1 kw-hr. for every 35 lb. of steam supplied. The initial steam pressure is 175 lb. per sq. in. gage and the steam is superheated 130 deg. fahr. The vacuum at the exhaust is 28 in. of mercury, barometer 29.58 in. Find the Rankine cycle efficiency, and the thermal efficiency.

3. Plot a complete expansion cycle, using absolute pressure as ordinates and energy available in B.t.u. as abscissas, for steam expanding adiabatically from an initial pressure of 250 lb. per sq. in. abs., and a steam temperature of 550 deg. fahr. to a back pressure of 1 in. of mercury abs. If the total energy available is to be divided into four equal parts, find the pressure drop for each part.

4. A steam turbine is to operate at a speed of 2400 r.p.m., and the mean diameter of the rotor is to be 40 in. The probable steam consumption will be 10,000 lb. per hr. with the supply steam at 250 lb. per sq. in. abs. and 150 deg. superheat, the condenser pressure to be 3 in. of mercury abs. Assuming adiabatic expansion and neglecting friction losses find:

(*a*) The number of impulse velocity-stages.
(*b*) The number of impulse pressure-stages.
(*c*) The number of impulse pressure-stages, if each has two velocity-stages.
(*d*) The number of pure reaction-stages.
(*e*) The number of impure reaction-stages.

5. A steam turbine has two velocity-stages. The steam is directed against the first row of blades with a velocity of 3500 ft. per sec. and at the rate of 7200 lb. per hr. The steam is discharged from this row at a velocity of 1900 ft. per sec. and is then directed upon the second row of blades from which it is discharged at a velocity of 200 ft. per sec. (*a*) Neglecting friction and other losses, how much horsepower will this turbine develop? (*b*) How much power would be developed if frictional effects reduce each of the velocities by 10 per cent? (*c*) Find the steam consumption in lb. per hp-hr. for part *b*.

6. The steam supply-pressure to a single-stage impulse turbine is 200 lb. per sq. in. abs. and the exhaust pressure 5 lb. per sq. in. abs., the steam being initially dry and saturated. The peripheral speed of the blades is 1200 ft. per sec. and the nozzle angle 20 deg. (*a*) Draw the velocity diagram determining the blade angle and the absolute exit velocity, neglecting friction and assuming adiabatic flow. (*b*) Also find the work done on the blades per second per pound of steam and the efficiency of the turbine.

7. A turbine is supplied with saturated steam at a pressure of 150 lb. per sq. in. abs. and the condenser pressure is 3 in. of mercury abs. Three pressure stages are to be used, and the intermediate pressures are to be such that in each of the first two stages 30 per cent of the work is delivered and in the last stage 40 per cent. Determine the approximate pressure range in each stage.

8. A reciprocating engine receives steam at 185 lb. per sq. in. abs. and a temperature of 400 deg. fahr., and exhausts at a pressure of 16 lb. abs. If a turbine is

added and the condenser pressure is 2 in. of mercury abs., what is the increase in the available heat?

9. Using Fig. 205, calculate the unit water rate of the 1000-kw. turbine when delivering 750 kw. and bleeding 25,000 lb. of steam per hr. Also find the heat content of the bled steam.

10. A turbo-generator delivers 300 kw. and uses 18 lb. of dry steam per kw-hr., steam pressure 160 lb. per sq. in. abs., and exhaust, 2 in. of mercury abs. The boiler supplying steam to the above turbine receives feedwater at a temperature of 102 deg. fahr. and the evaporation from and at 212 deg. fahr. is 9 lb. of water per pound of coal as received. Coal as received contains 11,800 B.t.u. per pound. What is the heat efficiency of the plant?

CHAPTER XVI

STEAM CONDENSERS

241. Foreword.—Theoretically, the lower the back pressure of an engine or turbine the greater will be the amount of work that can be obtained from a given weight of steam. By decreasing the exhaust pressure with unchanged cylinder volume, the area W_1 in Fig. 209 represents the work gained; whereas, using the steam in a turbine corresponds to a continuation of the expansion line as the exhaust pressure decreases and the gain is the work area W_2. It is therefore desirable to exhaust into a space in which the lowest possible pressure exists, when the work obtained per pound of steam is the only consideration.

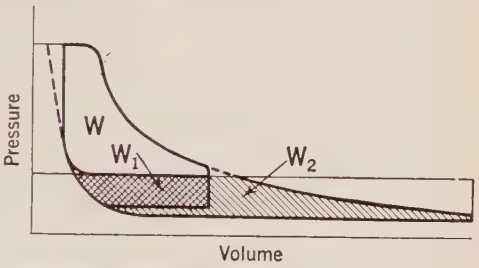


FIG. 209.—Diagram showing gain resulting from condensing operation.

A condenser is used in connection with steam engines and turbines for the purpose of obtaining one of two important results, and generally both of them: (1) a reduction of back pressure with the consequent increase in m.e.p. of the prime mover and (2) the recovery of the condensed steam which may be returned to the boiler as practically pure water. The condensation of steam in a closed vessel produces a partial vacuum due to the great reduction in volume of the vapor. A pound of dry steam at atmospheric pressure has a volume of approximately 26.8 cu. ft. Theoretically, if this quantity were contained in a steam-tight vessel of 26.8-cu. ft. volume at atmospheric pressure and condensation into water at 140 deg. fahr. occurs, the liquid will occupy only 0.0163 cu. ft. The volume of the liquid will be about 1/1650 of the space inside of the vessel and the absolute pressure will fall to about 3 lb. per sq. in. abs. These figures would be modified in actual practice by the presence of air and other non-condensable gases, and the pressure in the vessel would be considerably higher.

Steam condensers are devices in which the condensation of the

exhaust steam, and the removal of air and other non-condensable gases is a continuous process. A certain amount of air is present in water obtained from natural sources. Air may also leak into the condenser through the shell, the piping, the turbine casing and the various packings. The degree of vacuum (Art. 7) which can be obtained with condensing apparatus depends largely upon the tightness of the joints and the amount of air which enters with the steam as well as the temperature to which the condensed steam (condensate) can be reduced. When the pressure is reduced to below atmospheric the entrained air leaves the condensed steam in the surface condenser; air also leaves both the steam and the cooling water in the jet and barometric types of condensers. In any case the air remains in the condenser until positively removed.

Experience has shown that the highest practical vacuum suitable for ordinary reciprocating steam engines is 26 in. of mercury (30-in. barometer). Higher vacua are not economical. In the reciprocating engine full advantage of higher vacua cannot be secured by expanding steam to the condenser pressure, on account of the excessively large volumes of steam produced. Attempts to do so largely increase the bulk of a single cylinder or diminish the mechanical power of the engine, due to the early cut-off required. If expansion of the steam to pressures less than 2 lb. per sq. in. abs. (26-in. vacuum), were practical, the low-temperature exhaust steam would cool the cylinder walls to cause a very rapid increase in cylinder condensation.

For steam turbines the average practice is to employ a vacuum of 28 in. of mercury referred to a 30-in. barometer. Many installations under 1000-kw. operate on 26 to 27 in. and others of larger capacity exhaust to $28\frac{1}{2}$ to 29 in. of vacuum. There is a tendency on the part of steam-turbine manufacturers, because of the obvious advantages of steam turbines over reciprocating engines for operation at high vacua to call attention to the reduction in steam consumption when a plant employs a high vacuum. In obtaining this economy consideration must be given to the greater first cost of the condenser, pumps and piping, together with the increased operating expense. Local conditions, particularly the amount and temperature of the cooling water, determine in part the feasibility of a condenser where condensing operation is desirable. In locations which are practically at sea level and where there is an adequate supply of low-temperature cooling water, it would probably be profitable to install condensing apparatus of sufficient size to operate steam turbines at $28\frac{1}{2}$ to 29 in. of vacuum.

242. Classification of Condensers.—Condensers used in steam plants may be divided into two general classes: (1) *Jet condensers*, in which condensation of the exhaust steam is effected by direct contact between

the cooling water and the vapor; and (2) *surface condensers*, wherein the exhaust steam and the cooling medium do not come in direct contact with each other, the heat being extracted from the steam by conduction through metal walls.

By far the largest number of condensers built today are of the surface type. This type is used exclusively for marine purposes, as the recovery of the condensate is essential. For large steam-turbine installations, the use of jet condensers is exceptional, as apart from the recovery of the condensate, the auxiliary power consumption of a jet-condenser plant and the first cost of the requisite air pumps to attain the extremely high vacua desired, represent serious drawbacks. However, with moderate-size turbine units and for reciprocating steam engines, the jet type has a large field of useful application, especially where an abundant supply of good feedwater is available.

Surface condensers generally consist of a cast-iron or mild steel shell, fitted with a tube plate at each end. A great number of tubes extend between the tube plates, thus forming the cooling surface. In most cases steam is admitted to the condenser through an opening in the top of the shell, and water is passed through the tubes. Where the condenser is used in conjunction with a pumping engine, the reverse arrangement is generally adopted, that is, the water passes over the outside of the tubes and the steam is allowed to pass through the tubes. The latter arrangement is called a *water-works condenser*. Another form of surface condenser, known as an *evaporative condenser*, has the shell omitted. The steam is fed to the condenser tubes over whose outer surface water is sprayed. The cooling is effected mainly by evaporation of the water in the surrounding atmosphere.

Jet condensers may be classified as low-level jet and barometric condensers. Both types are similar in the manner in which the exhaust steam and cooling water are brought together, but differ in the method of removing the water and the condensate. In the low-level jet condenser the condensing chamber is at a low elevation, water being removed by a pump, and the overall height of the unit is low enough so that the condenser may be directly beneath the turbine or engine. With the barometric type, the condenser is placed at a sufficiently high level (about 40 ft.) above the point of water discharge to enable the water to drain away by gravity through a water-sealed " tail pipe " or barometric column.

243. Low-Level Jet Condensers.—Fig. 210 shows a low-level jet condenser of the parallel flow design, that is, the steam, cooling water and non-condensable gases flow in the same direction. The inlet cooling water enters an annular chamber at the top and passes into the steam

space through suitably designed nozzles. These nozzles break up the water into a finely divided state, thus insuring intimate contact with the incoming steam. The mixture of cooling water and condensate passes through a cone in which the air is entrapped in the water as far as possible. The mixture then falls in a solid stream through the lower part of the condenser body into the pump opening. The water is lifted from the source of supply to the nozzle injection head by means of the vacuum within the condenser, and removed by a centrifugal pump working against the vacuum in the condenser. The surplus air-vapor mixture at the lowest temperature is withdrawn from near the top of the condenser body into the suction inlet of an air pump or air ejector. In order to prevent the water from rising to an excessively high level in the condenser, a vacuum breaker is provided. This breaker consists of a float-controlled valve of generous size. When the water level reaches a predetermined height, the valve opens and admits air, diminishing the vacuum so that less water is drawn into the condenser. The vacuum breaker is not in action during normal operation, but is provided as a safety device in case the motor driving the pumps should stop, thus failing to remove the accumulated water, or in case the water injection valve is improperly adjusted.

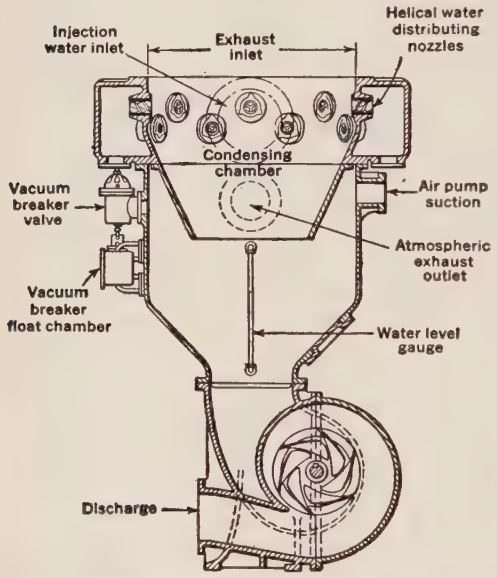


Fig. 210.—Section of Wheeler low-level jet condenser.

Jet condensers in which the cooling water, condensate and non-condensable gases are withdrawn by a single pump are called low-level, low-vacuum jet condensers because of the limited air capacity of the pump. In the condenser illustrated by Fig. 210, the non-condensable gases are removed by a separate pump or ejector, higher vacua are obtainable and the condenser is known as a low-level high-vacuum jet condenser. Under certain conditions the air and water can be removed by the kinetic action of the steam, in which case the jet condenser is designated as an *ejector* or *siphon condenser*.

The most widely adopted type of ejector condenser is the low-level *multi-jet condenser*, illustrated by Fig. 211. The condenser consists of a closed cylindrical chamber, in the upper end of which is a water nozzle case. The latter is fitted with a Venturi-shaped combining tube, made up of several sets of tapering rings joined together by equally spaced ribs. The combining tube is connected with a tail pipe, the lower end of which is immersed in water in a well or reservoir. The water-nozzle case is provided with a nozzle plate containing the water nozzles. The condenser body has two steam inlets, either one of which can be used. In Fig. 211 the side steam inlet is closed by a flanged cover.

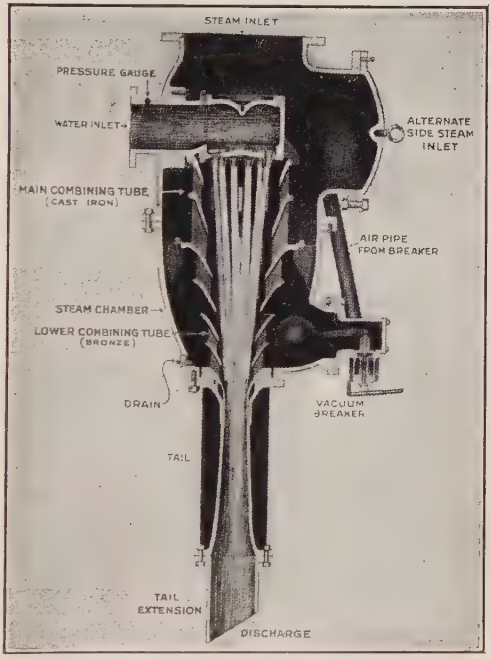


FIG. 211.—Section of Ingersoll-Rand low-level multi-jet condenser.

The injection water, Fig. 211, is forced under pressure into the water-nozzle case of the condenser and passed through the water nozzles, which are designed to discharge a specified amount of water according to the conditions of flow determined by the water pressure and the existing vacuum. As the water jets emerge from the nozzles they are directed into the throat of the combining tube, where they unite to form a single jet. The exhaust steam enters the condenser chamber through the steam inlet, flows through the annular passages between the tapering rings, comes into direct contact with the converging water jets and is condensed. Due to the combined effect of the external water pressure, the vacuum existing in the condenser, and gravity, the multiple water jets attain a velocity sufficient to entrain the condensed steam, air and non-condensable gases, and to discharge them into the hot well against the pressure of the atmosphere. The water jets create a vacuum by condensing the steam and maintain the vacuum by entraining and removing the air and non-condensable gases. No air pump or water removal pump is required.

A larger amount of water is required, as it is not practicable to subdivide the water jets in the ejector condenser to the same extent as in the other forms of jet condensers, but the absence of vacuum pumps may offset this disadvantage. Multi-jet condensers maintain vacua up to 29 in., referred to a 30-in. barometer, with 60-deg. fahr. cooling water and are suitable for turbine installations up to 10,000-kw. capacity.

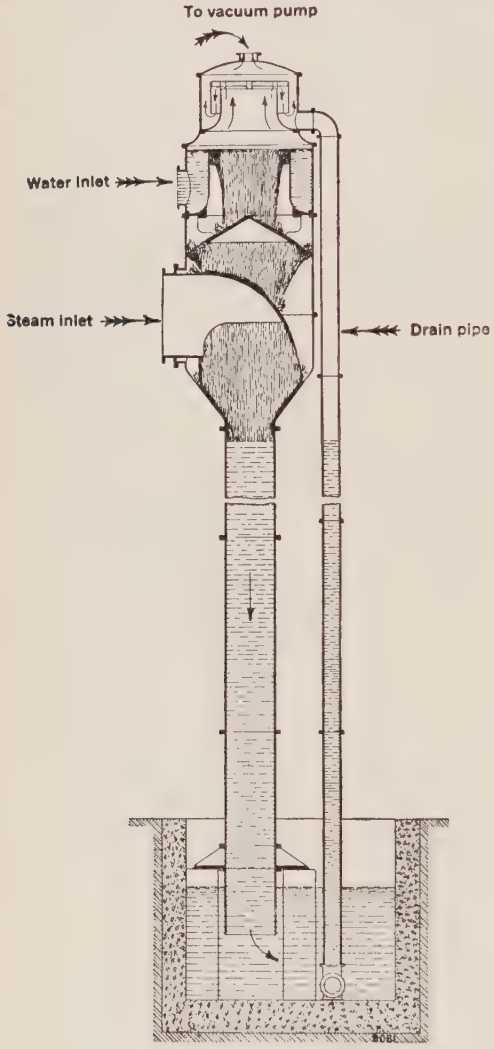


Fig. 212.—Ingersoll-Rand barometric condenser.

244. Barometric Condensers.—This type of jet condenser consists essentially of a head, similar to that used with the low-level jet condenser, and a tail pipe or barometric column which partly or wholly takes the place of the removal pump. Fig. 212 shows a section through an Ingersoll-Rand barometric condenser of the counter-flow type. In the condenser illustrated the water enters a pool in the upper part of the condensing head at a point above the steam inlet. Ample opportunity is given for the air content of the water to be removed by the vacuum pump. The water partially relieved of the air then meets the entering steam, and absorbs the latent heat. The air released from the steam passes upward through the falling sheets of water to a point above the water pool, and is removed by the vacuum pump.

The hot water resulting from the mixing process falls to the bottom of the condenser and the air at the extreme top is cooled to practically the temperature of the inlet water. This cooling of the air causes condensation of the water vapor accompanying it and the condensed vapor collects in the space surrounding the air cooler and flows down into the drain pipe. The vacuum pump, usually called a *dry-air pump* or *dry-vacuum pump*, handles cool gases containing little vapor and practically no water. The hot water flows downward into the tail pipe, which is about 35 ft. in length. The lower end of the tail pipe dips into the hot well. As atmospheric pressure can support a column of water about 34 ft. high, theoretically, the tail pipe forms an automatic removal pump; water flowing from the tail pipe as rapidly as it accumulates.

In both low-level and barometric condensers the custom is to lift the water from the source of supply to the injection head by means of the vacuum within the condenser. The greatest height to which the water can be lifted by this means is about 18 ft. with a 28-in. vacuum. Where necessary a suitable type of pump is used to aid in lifting the water to the condenser head. The construction of the barometric condenser is simple, there being no moving parts, no nozzles and contracted orifices to become clogged and no need for vacuum breakers. Barometric condensers require a long exhaust pipe because of the long tail pipe. The following disadvantages result: (1) greater cost of piping, (2) the possibility of air leakage, and (3) the increased resistance offered to the flow of exhaust steam.

245. Surface Condensers.—The surface condenser does not permit the cooling medium and the exhaust steam to come into contact with each other. The device consists essentially of a cylindrical or rectangular vessel through which pass numerous brass or alloy tubes, Fig. 213. In the usual surface condenser, the cooling water circulates through the tubes and the steam to be condensed is brought in contact with the outer surfaces of the tubes. Condensers of this type may be " single-pass " in which the water flows in one direction only through all the tubes, or " two-pass " in which the water flows in one direction through part of the tubes and then returns through the remainder of the tubes. Surface condensers may operate on the wet vacuum or the wet and dry vacuum principle. In the case of wet vacuum operation, one pump (wet-air pump) connected to the bottom of the condenser chamber removes the condensate and along with it the larger volume of air and non-condensable gases. With the wet and dry system the condensate is removed from the lowest point by one pump and the air at some point higher up in the condenser by a dry-air pump.

A surface condenser and its auxiliaries should accomplish the follow-

ing results: (1) *Steam* should enter the condenser with the least possible resistance, and the pressure drop through the condenser should be reduced to a minimum. (2) *Air*, a poor conductor of heat, should be rapidly removed from the heat-transmitting surfaces. The air should be collected at suitable places, practically free from entrained water vapor and cooled to a low temperature. The air removal should be with a minimum volume and the least expenditure of mechanical energy. (3) *Condensate* should also be rapidly cleared from the heat-transmitting surfaces, freed from air, collected at suitable points for removal and returned to the boiler at the maximum practical temperature. (4) *Circulating water* should pass through the condenser with little friction,

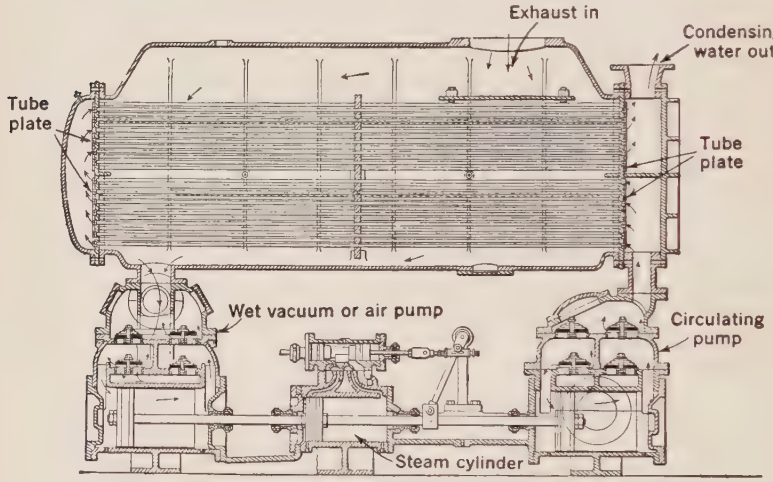


Fig. 213.—Wheeler-Admiralty surface condenser and equipment.

with a minimum amount of deposited materials and with a maximum amount of heat absorption.

One form of surface condenser, mounted over a combined circulating water and wet-air pump of the piston type, is shown by Fig. 213. The exhaust steam enters at the top of the rectangular shell and passes downward over the water-cooled tubes. A baffle plate is located under the exhaust inlet in order to distribute the steam properly over the tubes and to prevent erosion of the upper row of pipes. The condenser tubes are usually made of ordinary brass (62 copper, 38 zinc) or Muntz metal for pure water and Admiralty brass (70 copper, 29 zinc, 1 tin) for salt water and impure river waters. The outside diameter of the tubes varies from $\frac{5}{8}$ to 1 in. The condensate mixed with gases and vapors is removed from the bottom of the condenser by the wet-vacuum pump and

discharged to the hot well. The cooling water is forced through the tubes of the condenser by means of the reciprocating circulating-pump. The water enters the lower tubes at the right end, makes two " passes " through the condenser and leaves at the top. The vertical support plate at the center of the condenser is usually made of cast iron and in long-tube types is so spaced as to allow not more than 6 ft. of unsupported tube. This type, Fig. 213, of condenser equipment is intended for steam engines and small turbines where the vacuum desired does not exceed 26 in. of mercury.

With the arrangement illustrated by Fig. 213, the steam which condenses upon the upper tubes falls as rain from tube to tube until it settles at the bottom. The outer surfaces of the lower tubes are, therefore, practically covered with water which results in two disadvantages. First, these tubes carry the coolest circulating water and cool the condensate coming in contact with them while the water flowing through the tubes is unnecessarily heated. Cooling of the condensate gives a lower hot-well temperature than would otherwise be obtained. If the condensate is to be used for boiler feedwater, the temperature in the hot well should be maintained as high as possible, since this water must again be heated to boiler temperature with a corresponding expenditure of heat. Second, tubes which are being used to cool water covering them are of little value as condensing surface, so that such surface in a condenser is practically useless.

More efficient operation can be obtained by removing the condensate as rapidly as formed, thus allowing the entering water to act on the final cooling surface for the non-condensed gases just before they enter the vacuum pump. A condenser of improved design passes the condensate through a set of tubes so located that the engine exhaust passes over them before impinging on any tubes carrying cooling water. Partial condensation of the exhaust steam results and the temperature of the condensate within the tubes is raised to approach that of the exhaust temperature of the engine.

When surface condensers are operated with a vacuum less than 26 in. of mercury, wet-vacuum pumps are generally used, but the best types of dry-air pumps must be used with well-designed condensers, when higher vacua are desired. An installation of a surface condenser with the necessary auxiliaries as applied to a steam turbine of moderate size is shown by Fig. 214. The condensate is removed by a centrifugal hot-well pump, P_c. The cooling water is supplied by the centrifugal (circulating) pump, P_w. The non-condensable gases are removed near the bottom of the condenser by means of a rotative dry-vacuum pump P_a. The dry-vacuum pump shown is merely a reciprocating air com-

pressor with very small clearance which operates between a high vacuum and atmospheric pressure. The exhaust steam passing to the condenser may be bypassed through a valve to the atmosphere through pipe *A*, so that the turbine may operate non-condensing if the condenser is out of service. The condenser body is flexibly connected to the turbine by means of a corrugated copper expansion joint. When no expansion joint is provided between the turbine and condenser, the condenser may be mounted on spring supports to take care of expansion and contraction.

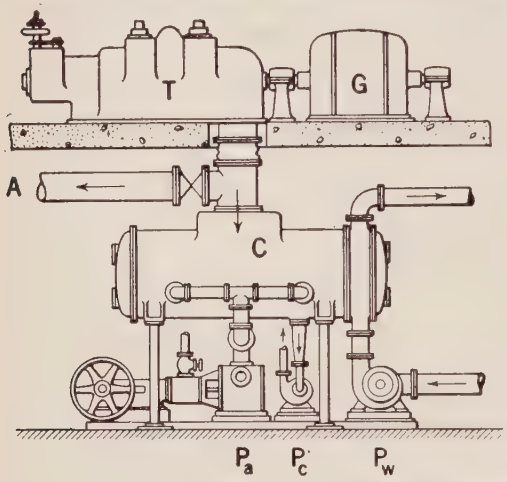


FIG. 214.—Surface condenser and auxiliaries, as applied to a steam turbine of moderate size.

246. Vacuum Pumps.—The vacuum or air pumps used in connection with steam-condensing plants are of four distinctive types, (1) reciprocating, (2) rotary positive displacement, (3) hydraulic entrainment, and (4) steam ejector. The earliest dry-vacuum pumps were merely air compressors which worked between a high vacuum and atmospheric pressure. These pumps receive "air" at the pressure existing in the condenser and compress it to a pressure high enough to discharge it to the atmosphere. Some of these reciprocating pumps are arranged with a crank and flywheel and are known as rotative dry-vacuum pumps. Leakage, defective jacket cooling and re-expansion of the air in the clearance spaces make it difficult to maintain a vacuum over 26 in. of mercury. Another disadvantage of reciprocating air pumps is the excessive size and high cost when made in large capacity. Rotary positive-displacement vacuum pumps are similar in construction to the blower shown by Fig. 231, in which air from the condenser is drawn through the intake and discharged to the atmosphere.

An example of a hydraulic air-pump, consisting primarily of a multivane wheel in conjunction with an ejector is shown by Fig. 215. Water enters the central chamber of the pump from which it is discharged through the distributor *N*. The water is caught up by the blades *V* of the wheel, rotating at a suitable speed. The blades divide the water

into a succession of small elements which are impelled down the diffuser pipe N' in the form of sheets. These sheets form a series of water pistons and the air entering at the top is trapped between successive sheets and carried out against the atmospheric pressure. The Leblanc pump is a positive displacement pump, in which water pistons are used. In starting, water or steam under the necessary pressure can be admitted at P, thus creating sufficient vacuum to establish the flow of water and air. The hydraulic air-pump requires more power than the reciprocating pump but it can maintain a slightly higher vacuum. However, the reciprocating pump can establish a vacuum much quicker than the hydraulic pump.

The steam-ejector type of vacuum pump has practically supplanted other types for steam condensers, because of its compactness, absence of moving parts, simplicity of operation and high efficiency. One or more steam jets moving at a high velocity entrain the noncondensable gases and the mixture passes through an expanding tube in which the velocity is reduced with a corresponding increase of pressure. The pressure rises to a value sufficiently above atmospheric to make possible the discharge of the mixture to the atmosphere.

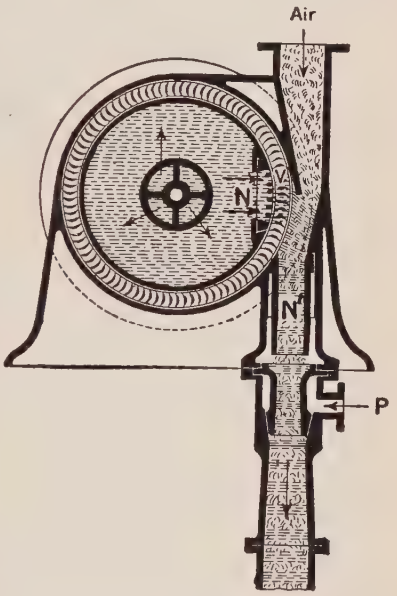


Fig. 215.—Westinghouse-Leblanc air pump.

The C. H. Wheeler Radojet vacuum pump, Fig. 216, is an ejector combination using a tubular ejector in the first stage and a radial flow ejector in the second stage. Live steam enters through opening L, flows through strainer cage 1, pipe 2, auxiliary steam valve 3, strainer cage 4, to the first stage nozzles at 5. The steam expands in the nozzles, leaving with a very high velocity and while passing across suction chamber 6, the primary jet withdraws the saturated air from the condenser through the suction opening S. The mixture passes into the diffuser 7, from which it is discharged at a higher absolute pressure than that of the air entering at S, into a double passage 8, communicating with the suction chamber 9 of the second stage. The two suction chambers are annular thus giving the commingled fluid a large entrainment surface.

Steam is simultaneously delivered through the strainer cage 1 into passage 10, which communicates with the annular expansion nozzle of the second stage formed between nozzle 11 and nozzle point 12. Nozzle point 12 may be adjusted towards or away from nozzle 11 by the adjusting screw 13, which forms a part of 12, to vary the cross-section of the nozzle passage thereby changing the expansion ratio of the steam. The steam delivered radially to the second stage by the annular nozzle 11 expands between 11 and nozzle point 12, leaving it as a jet at high velocity in the form of an annular sheet. This sheet entrains the mixture discharged from the first stage and carries it into the annular diffuser 14, thereby compressing the mixture to atmospheric pressure and discharging it into casing 15 to discharge opening *D*. The radial flow of the steam after passing the nozzle point 12 gives the device its trade name " Radojet."

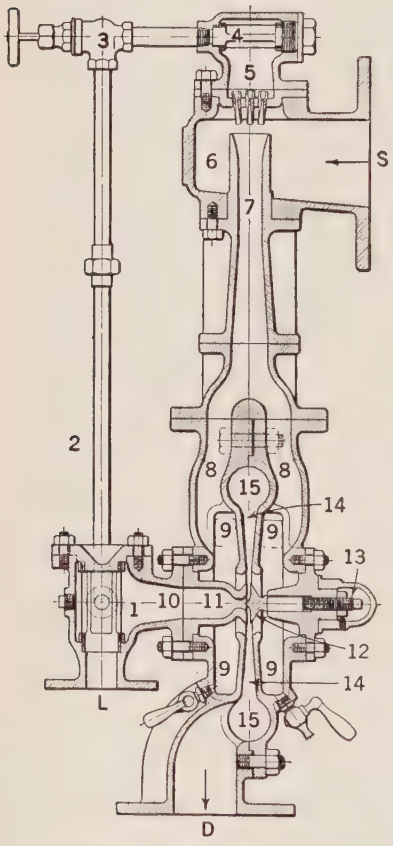


Fig. 216.—Radojet vacuum pump.

247. Cooling Water Required for Condensers.—Isolated attempts have been made to dispose of the heat rejected in a condenser by passing air over the cooling surface. The power consumption involved in attaining the necessary cooling effect rules the air-cooling method out of practical consideration except in locomotive practice. The only effective means of providing the necessary cooling medium is by direct heating of water, or as in the case of condensers of the evaporative type by the evaporation of water in immediate contact with the outside atmosphere. The weight of cooling water required varies with the type of condenser and with the conditions of operation, such as initial temperature of water, vacuum desired, etc. The quantity of water required can be determined approximately by calculation and the values thus found must then be increased by such factors as experience has shown to be necessary.

In *jet condensers* the water and condensate are discharged as a mixture and therefore have the same average discharge temperature. Neglecting the effect of radiation, leakage and the entrained air, the heat given up by the steam is equal to the heat received by the cooling water.

$$W_s[H - (t_2 - 32)] = W_w(t_2 - t_1)$$

or

$$W_w = \frac{W_s[H - (t_2 - 32)]}{t_2 - t_1} \quad . \quad . \quad . \quad . \quad . \quad (142)$$

in which W_w = weight of cooling water required, lb. per hr.
W_s = weight of steam condensed, lb. per hr.
H = total heat of the exhaust steam, B.t.u. per lb.
t_2 = final temperature of mixed cooling water and condensate, deg. fahr.
t_1 = initial temperature of cooling water, deg. fahr.

The value of t_2 would be the same as that corresponding to the absolute pressure in the condenser if it were not for the pressure exerted by the non-condensable gases. In higher vacuum jet condensers the air pressure is low and the temperature t_2 will range from 6 to 10 deg. below that of the steam at condenser pressure. In the usual low-vacuum condenser t_2 may be 15 to 25 deg. less than the steam temperature. The weight of cooling water required per pound of steam will vary from 20 to 60 lb. depending mainly upon the initial temperature of the water and the vacuum desired. Experience has shown that 10 per cent or more should be added to the calculated values of W_w from equation 142 to obtain the weight which would probably be used.

For *surface condensers*, the heat given up by the exhaust steam equals the heat absorbed by the cooling water, neglecting the effect of radiation, leakage and entrained air.

$$W_s[H - (t_c - 32)] = W_w(t_2 - t_1)$$

or

$$W_w = \frac{W_s[H - (t_c - 32)]}{t_2 - t_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (143)$$

in which t_c = temperature of the condensate, deg. fahr.
t_2 = final temperature of the cooling water, deg. fahr.
Other notations as in equation 142.

The discharge temperature t_2 of the cooling water is generally 10 to 25 degrees lower than the temperature corresponding to the steam pressure in the condenser. The temperature of the condensate t_c (hot-well tem-

perature) is often 15 or more degrees below that of the exhaust steam, but good design allows t_c to approximate very closely the temperature corresponding to the total pressure in the condenser. The weight of cooling water required per pound of steam will vary from 25 to 50 lb. with low vacua and 50 to 100 lb. for vacua over 27 in. of mercury. For engine condensers, 1 sq. ft. of cooling surface is generally allowed for every 10 lb. of steam condensed per hour for 24 to 26 in. vacuum with 70 deg. fahr. cooling water. An average figure for turbine condensers using high vacua is 6 lb. of steam condensed per hour per square foot of tube surface. In some of the recent central station installations, surface condensers of the vertical single-pass type are employed; the surface used varies from 0.75 to 1.25 sq. ft. per kw. of turbine rating. For the horizontal single- and two-pass types, about 1 to 2 sq. ft. per kw. are required.

248. Comparison of Condensers.—Jet condensers do not depreciate as rapidly and have a smaller initial cost than surface condensers. The tubes of surface condensers frequently show considerable corrosion in short intervals of time. Large quantities of dissolved gases enter the condensing space of a jet condenser with the cooling water and the liberation of these gases under the reduced pressure tends to decrease the vacuum, that is, increase the exhaust pressure. The additional work required to remove the liberated non-condensable gases may partly offset the advantage of lower first cost of the jet condenser.

Surface condensers are installed when high vacua are to be maintained and where the re-use of the condensate as boiler feedwater is essential. The heat conductivity of the tubes in a surface condenser may be decreased by lubricating oil deposits from the exhaust steam of a reciprocating engine or the deposit of foreign matter from the cooling water. The use of acidulated water or water containing foreign matter in large quantities may preclude the use of surface condensers.

The barometric condenser proper is not costly, but the piping to it is expensive and adds to the possibility of air leaks. Danger of flooding the turbine does not exist as in the case of a low-level jet condenser. The use of acidulated water is possible and the maintenance is low as compared with the surface condenser in which the tubes require frequent cleaning in order to obtain high vacua. The equipment for low-level jet and barometric condensers requires no hot-well pumps and is simpler than for surface condensers. Low-level jet and barometric condensers also require less building space and less circulating water than surface condensers of equal capacity.

249. Water-Cooling Methods.—When an ample supply of water is unobtainable at a low cost or from a natural source, such as a lake or

river, some means of cooling the circulating water may be used. The condensing water is often circulated continuously, being cooled after each passage through the condenser. The cooling is generally effected by the exposure of a large surface of the water to the air. The resultant evaporation of some of the water with the absorption of the latent heat of vaporization cools the remainder, so that the cooled water can be used again. This process of recooling the water is usually accomplished by means of cooling ponds or cooling towers.

The oldest means of cooling and storing condensing water is to discharge the hot water into a pond of sufficient area, so that the water is cooled by the air passing over the surface of the pond. This method requires ponds of large size, and in order to reduce the area necessary the water may be sprayed into the air over the pond, thus increasing the rate of cooling. The cooling action is effected by convection and evaporation. Natural ponds without spray nozzles require about fifty times the area of spray-cooling systems.

Because of space requirements and other objectionable features inherent in the spray pond, the cooling tower has been developed. Cooling towers are large wood or metal structures generally arranged with some form of baffling devices. The hot water is delivered to the top and falls in thin sheets or sprays into a tank below. Air enters at the bottom and flows upward, cooling the water by contact (convection) and by the partial evaporation which results. The circulation of air may be effected by either natural or forced draft. With natural draft, the difference of temperature between the air inside and outside the tower causes the flow, the chimney action is aided by fitting a stack to the top of the tower. Forced-draft towers have entirely closed sides, except for openings at the base containing one, two or more fans; the heated and saturated air leaves through the open top of the tower.

PROBLEMS

1. Find the quantity of cooling water theoretically required per pound of dry steam condensed in a jet condenser in which a vacuum of 25 in. Hg is maintained, when the barometer reads 29.5 in. Hg. The initial temperature of the water is 70 deg. fahr.

2. A surface condenser receives circulating water at a temperature of 56 deg. fahr. and discharges it at a temperature of 72 deg. fahr. A vacuum of 28 in. Hg is maintained with the barometer at 29.4 in., and the temperature of the condensate discharged to the hot well is 78 deg. fahr. Find the quantity of circulating water theoretically required, if the entering steam contains 10 per cent moisture.

3. A steam engine delivers 100 brake hp. operating on dry steam at 150 lb. per sq. in. abs. pressure and exhausts into a jet condenser having a vacuum of 24 in. Hg. Barometer registers 29.25 in. Hg. How much 70 deg. fahr. cooling water would

theoretically be required in gallons per minute, if the engine has an economy of 25 lb. of dry steam per brake hp-hr.? Mechanical efficiency, 90 per cent.

4. A 10,000-kw. steam turbine uses 14 lb. of steam per kw-hr. The steam is supplied to the turbine at 350 lb. per sq. in. abs. and a total temperature of 550 deg. fahr. The steam is exhausted into a surface condenser against an absolute back pressure of 1.5 in. Hg. Temperature of cooling water entering condenser 70 deg. fahr., temperature of cooling water leaving condenser 120 deg. fahr. and temperature of condensate 90 deg. fahr. Find the capacity of the circulating pump in gallons per minute, assuming no losses and adiabatic expansion in the turbine.

5. In a jet condenser, the initial temperature of the cooling water is 55 deg. fahr. and the final temperature 105 deg. fahr. Dry steam enters an engine at 140 lb. per sq. in. abs. and the engine exhausts into the condenser at 2 lb. per sq. in. abs. The engine develops 300 i.hp. and uses 20 lb. of steam per i.hp.-hr. (*a*) Find the theoretical gallons of condensing water used per hour. (*b*) What is the theoretical quality of the steam entering the condenser? (NOTE.—Work is done in the engine by part of the heat in the steam.) (*c*) Find the thermal efficiency of the engine, based on i.hp.

6. A surface condenser uses 50 lb. of 60 deg. cooling water per pound of steam. Absolute pressure in condenser is 3 in. abs. and the condensate temperature is 5 deg. below that of the steam in the condenser. Find the probable temperature of the cooling water leaving the condenser. Assume quality of entering steam as 85 per cent.

7. A 15,000-kw. turbine uses 12 lb. of steam per kw-hr. at rated load. Initial steam pressure 300 lb. abs., superheat 250 deg. fahr. and the vacuum in a surface condenser is 29 in. referred to a 30-in. barometer. Temperature of the inlet circulating water is 57 deg. fahr. Outlet circulating water is at 70 deg. fahr. and condensate 3 deg. less than the steam temperature. Assuming the efficiency of the turbo-generator as 95 per cent and a mean value of the coefficient of heat transmission as 450 B.t.u. per hr. per deg. fahr. difference in temperature per sq. ft.

Required:

(*a*) Weight of cooling water per hour.

(*b*) Sq. ft. condenser tube surface.

(*c*) Number of 1-in. diameter tubes, 0.049 in. thick, if the water velocity through tubes is 6 ft. per sec. Use a two-pass condenser.

(*d*) Length of the condenser tubes.

CHAPTER XVII

PUMPS

250. Classification.—Pumps, for imparting velocity and pressure to fluids, may be grouped as follows:

1. Piston pumps { Single cylinder / Multi-cylinder
2. Centrifugal pumps
3. Rotary pumps
4. Fluid-impellent pumps { Continuous flow / Intermittent flow

Piston pumps work by exerting a pressure directly on the fluid being pumped. Centrifugal pumps exert the pressure indirectly by gradually arresting the momentum of the fluid given to it by rapid rotation of an impeller within a suitable casing. The water inlet and discharge of the piston pump are controlled by valves which open and close intermittently, whereas, in the centrifugal pump the entry and discharge are continuous with no valves and no (pump) controlling devices. Rotary pumps, similar to Fig. 231, combine the positive-pressure method of the piston pump with the continuous entry and discharge of the centrifugal pump by employing rotating impellers (or screws).

In fluid-impellent pumps the velocity and pressure of one fluid act directly on another, thereby imparting all or part of its energy to the latter fluid. Injectors (Art. 100), ejectors, aspirators, and some air lifts are of the continuous-flow type. The pulsometer pump, hydraulic ram, pneumatic pump, and the gas-explosion pump, are typical intermittent fluid-impellent pumps. These special forms of pumps have their own more or less definite applications and their use is generally dependent upon special service conditions. Fluid-impellent pumps will not be further discussed as they are not usually in commercial competition with other forms of pumps.

Vacuum pumps for the removal of air and vapors from steam condensers have been described in Chapter XVI.

251. Direct-Acting Steam-Driven Piston Pump.—A piston or reciprocating pump in its simplest form consists of a cylinder provided with

inlet and outlet openings, suitably controlled by valves, and a reciprocating piston or plunger working within the cylinder. This type of pump may be used for forcing feed water into a boiler, removal of condensate and air from a condenser, and in elevator and water-works service.

A steam pump that has no flywheel is known as a *direct-acting pump*, Fig. 219; the water piston and steam piston are fastened to the same piston rod and the steam pressure is transmitted directly to the water. In this type of pump the steam is not used expansively, and therefore the steam consumption is large. When a single steam cylinder and a single water cylinder are employed the pump is called a simple or *simplex steam-driven pump;* when two steam cylinders and two water cylinders are used, the pump is called a *duplex steam-driven pump.* The size of this type of pump is stated by giving first the diameter of the steam cylinder, next the diameter of the water cylinder, and then the length of the stroke; thus 10 in. × 6 in. × 12 in.

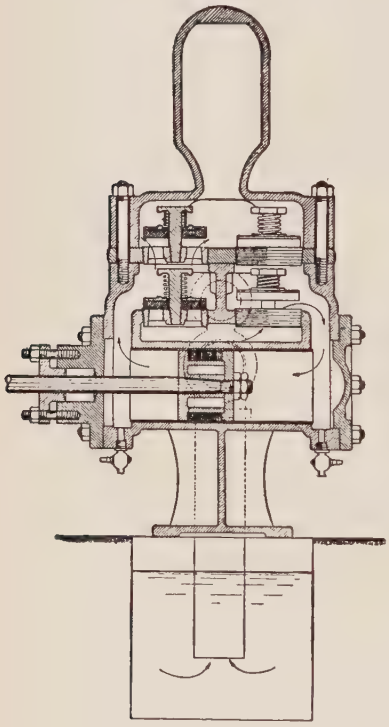


Fig. 217.—Water end of a direct-acting pump.

Direct-acting steam pumps for power-plant service are of the reciprocating double-acting type. Such pumps are designed to raise water by suction from a lower level, and to deliver the water during each stroke of the piston or plunger to a boiler, tank, etc. The movement of the piston toward the left, Fig. 217, forces the water through the discharge valve (upper left), as indicated. Coincidentally, a partial vacuum is created on the right side of the piston and when the reduction in pressure is large enough, atmospheric pressure forces water up the suction pipe and through the suction valve (lower right), as shown. On the return stroke of the piston the right side discharges water and the left side receives water from the source of supply. The intake water often flows under pressure to the suction pipe, as from an elevated tank, lake, or from street mains.

252. Simple Steam-Driven Pump.—The water end of the steam-driven single-cylinder pump, Fig. 218, has the same general construction as shown by Fig. 217. The steam cylinder and valve chest are practically the same as those described under the steam engine, with the addition of an auxiliary control F of the slide-valve G. The piston C is driven by steam admitted under the slide-valve G. As G is shifted from right to left, it alternately connects opposite ends of the cylinder A with the live-steam pipe and the exhaust. Slide-valve G is actuated by the auxiliary piston F. The hollow ends of F are filled with steam and this

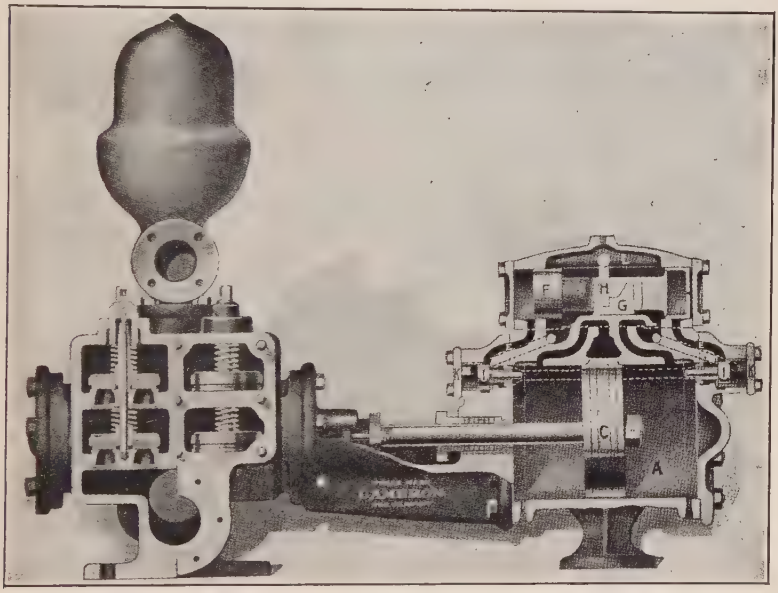


FIG. 218.—Section through simple direct-acting steam pump.

steam, issuing through a hole in each end, fills the spaces between the plunger and the heads of the steam chest. Piston valve F, under ordinary conditions, is balanced and motionless, the pressure being equal at each end. When the main piston C has traveled far enough to the left to strike and open the reverse valve I, the steam flows from the left end of F through port E to the exhaust. Because of this reduction in pressure at the left end of F, the piston valve F is forced to the left by the higher pressure at the right end; the movement of F actuates the slide-valve G, thus reversing the stroke of the piston. In its movement, F acts as a slide-valve to close the port E, and is cushioned by the confined steam between the port and steam chest cover. As soon as piston C moves out of contact with valve I, the latter is closed by steam pressure

conveyed directly from the steam chest through ports K, as shown by the dotted lines.

The *water valves* of piston pumps are generally of the poppet-disk type rising vertically from flat seats. The pump valves shown in Figs. 217 and 218 consist of a rubber composition disk supported by a brass cap and are held firmly to the seat by coil springs. For pressures exceeding 200-lb. gage, or for handling hot water, spring-loaded metallic valves are frequently used.

Air chambers attached to the discharge of piston pumps, as in Figs. 217 and 218, provide an elastic element in the pipe line, reduce the shocks and irregularities, and maintain a uniform flow of water in the pipe. The

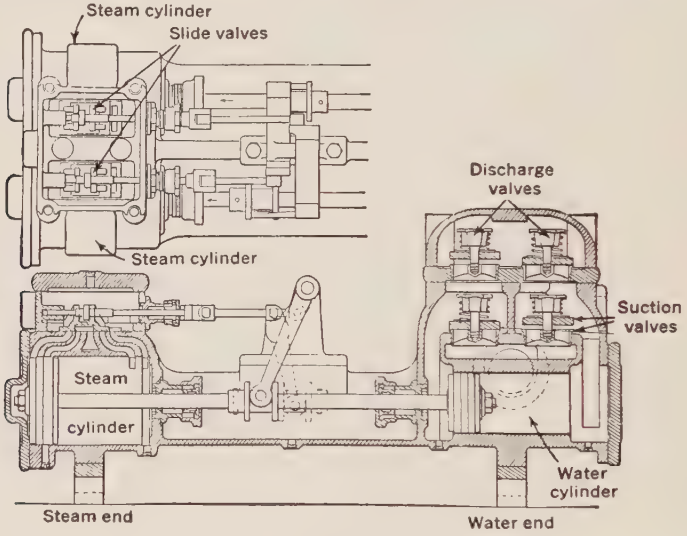


FIG. 219.—Duplex steam-driven pump.

air in the chamber is compressed during each stroke of the water piston; when the piston stops momentarily at the end of the stroke, expansion of the compressed air tends to maintain a uniform rate of discharge. Air chambers are a necessity for simplex pumps, single-acting pumps, and with crank and flywheel pumps. The air chamber may be omitted for duplex double-acting pumps (Fig. 219), for sizes 10 in. $\times$ 6 in. $\times$ 10 in., or less. Suction air chambers are frequently provided on long- or high-suction lifts for single pumps, on all fire pumps, and any pumps which run at high speeds, particularly those having a short stroke.

253. Duplex Steam-Driven Pump.—A duplex pump, Fig. 219, has two steam cylinders and two water cylinders. This pump comprises, in effect, two single pumps placed side by side; water entering through a

common suction pipe and discharging into a common discharge chamber and pipe. The operation of the water end of the duplex pump is similar to the operation of the water end of the direct-acting pump illustrated by Fig. 217. The steam valve of each cylinder is actuated by the piston rod of the other cylinder, through a system of oscillating rocker arms and bell-cranks. The flow of water is practically continuous as one or the other of the pistons is always in motion.

The slide valves used in the steam end of duplex pumps have no lap. Separate inlet and exhaust ports are provided at each end of the steam cylinder in order to decrease the valve travel and minimize steam leakage, also to cushion the piston near the end of the stroke with the entrapped exhaust steam. There is some lost motion between the slide-valve and its operating mechanism thus permitting adjustment of the piston stroke

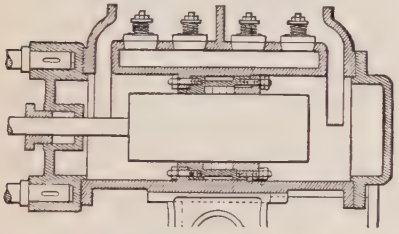


FIG. 220.—Pump plunger, inside-packed with fibrous hydraulic packing.

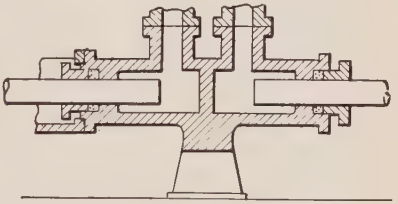


FIG. 221.—Outside end-packed plunger pump.

and preventing the pump from stopping in a position from which it cannot be started by admitting steam to the steam chest. This arrangement also causes a pause in the flow of water at the end of the stroke, thus giving the water valves an opportunity to seat quietly.

254. Plunger Pumps.—The water end of a direct-acting pump is often fitted with a plunger (Fig. 220) instead of a piston (Fig. 219). The plungers pass through fibrous or metal-packed stuffing boxes to prevent leakage, the packing being located either inside or outside of the cylinder. In Fig. 220, the packing is inside the cylinder, thus leakage is not readily detected. The packing is adjusted and replaced by removing the cylinder head. The plungers, Fig. 221, are outside packed at the ends of the cylinder; leakage is visible and may be prevented by adjusting the gland against the packing. These two plungers move together, being connected to an external yoke, the latter is usually actuated by a reciprocating steam engine. Plunger pumps are generally used for discharge pressures above 200 lb. per sq. in.

Piston and plunger pumps which operate at a uniform speed with the aid of the inertia effect of a flywheel are called *flywheel* or *rotative*

pumps; P_a, Fig. 214, illustrates a flywheel pump. When the reciprocating movement of the piston or plunger is derived from the rotation of a mechanically driven crankshaft, the pump is known as a *power pump.* Power pumps may be operated at constant speed by gears, belts or chains, and the source of power can be steam, gas or electricity. When one plunger is employed they are called simple or simplex; two plungers duplex; and three plungers, triplex power pumps.

Fig. 222 shows a vertical triplex power pump (single-acting) driven by an electric motor through a set of gears. The rotation of the crank-

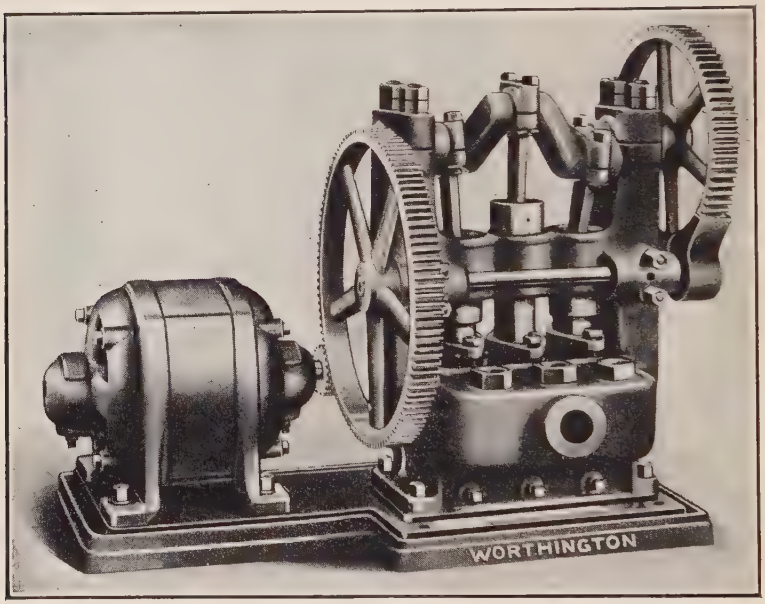


FIG. 222.—Single-acting triplex plunger pump.

shaft is transmitted to each plunger by a connecting rod. The base is made in compartments and the operation is the same as for the water end of any piston pump. The efficiency of this type of pump depends largely upon the character of the driving unit and the efficiency of the transmitting mechanism. With an electric motor drive the overall efficiency of power pumps is generally between 60 and 80 per cent.

255. Centrifugal Pumps.—The centrifugal pump is a simple mechanism with no valves or springs. In brief, the centrifugal pump consists of an impeller rotating within a casing, as shown in Figs. 223 and 224. Centrifugal pumps are relatively high-speed machines and are generally direct-connected to a steam turbine or an electric motor; belt and rope

drives, or internal-combustion engines may also be used. Water enters the impeller, Figs. 223 and 224, at the center (around the shaft), flows radially outward and leaves the periphery of the impeller at a velocity which is a resultant of the tip velocity of the impeller blade and the radial flow of the water through the pump. This velocity of the water is gradually decreased and the energy of motion is converted into pressure head in the pump casing in which the impeller rotates. The water is under pressure and will flow out of the pump against the resistance to be overcome. The design of a centrifugal pump is generally directed towards converting the energy of velocity into potential energy without loss, minimizing the friction of disk rotation, and balancing the end thrust of the impeller shaft.

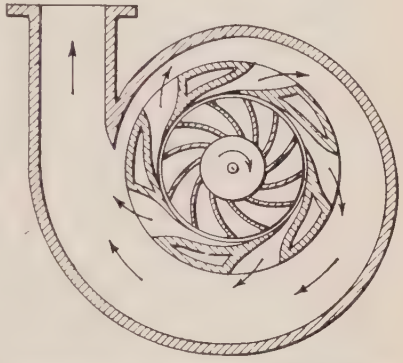


FIG. 223.—Single-stage turbine pump with volute casing.

Centrifugal pumps may be divided into the following groups: (1) *Turbine and volute;* (2) *number of stages;* (3) *single-suction and double-suction;* (4) *open-impeller and enclosed-impeller;* (5) *horizontal and vertical.*

The turbine centrifugal pump has an impeller which is surrounded by a casing containing diffusion vanes (stationary) as shown in Fig. 223.

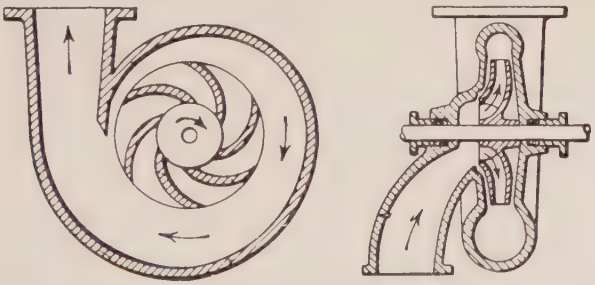


FIG. 224.—Single-stage, single-suction volute centrifugal pump.

These diffusion vanes provide gradually enlarging areas thus reducing the velocity of the water leaving the impeller and changing the velocity head into pressure head. The volute pump, Fig. 224, uses no diffusion vanes but the casing is of a spiral type forming a gradually increasing water space, thus gradually reducing the velocity of the water as it flows

from the impeller to the discharge pipe. Volute pumps ordinarily have but one impeller. The volute pump is considered superior to the turbine pump from a standpoint of first cost, simplicity and size but for discharge pressures greater than 50 lb. per sq. in., the turbine pump usually shows better performance.

The multi-stage centrifugal pump, Fig. 225, consists of two or more impellers through which the water passes successively. The impellers are mounted on the same shaft and revolve in separate compartments or stages. The discharge from each stage is delivered to the suction of the next higher stage, the final delivery pressure depending upon the number of stages. Multi-stage centrifugal pumps are generally employed when

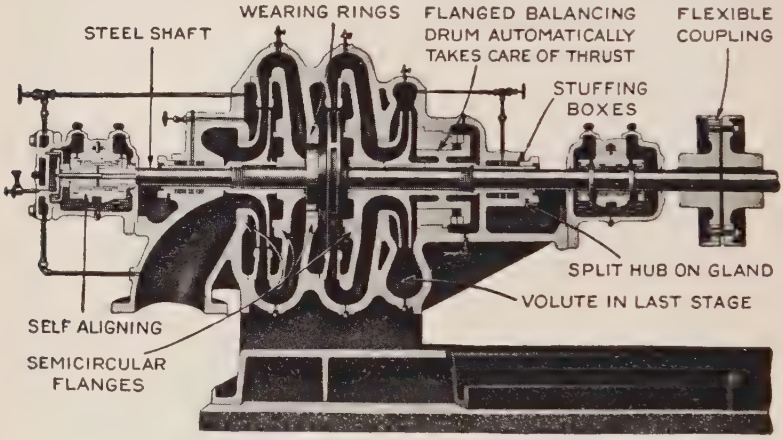


Fig. 225.—Cameron three-stage centrifugal pump.

the discharge pressure exceeds 100 lb. per sq. in. These pumps are usually of the turbine type with single inlet impellers. In Fig. 225 the first two stages are of the turbine type and the third stage employs a volute impeller.

In the single-suction centrifugal pump, Figs. 224 and 225, the water enters the impeller from one side only. A double-suction pump, Fig. 226, is one in which the water enters from both sides. For a given impeller diameter, the double-suction pump will discharge more water than a single-suction pump. The side thrust produced by the pressure of the water as it enters the one side of the impeller in the single-suction pump necessitates special provision for counterbalancing this thrust. The counterbalancing may be accomplished by a thrust bearing or by an equalization of pressures (hydraulic balancing) on the two sides of the rotor.

The open-impeller centrifugal pump, similar to Fig. 224, is sometimes called a fan pump. The operation is similar to that of a paddle-wheel revolving in a casing. Poor water-guidance results in excessive eddying of the water, also a large amount of water escapes between the blades of the impeller and the casing walls. The open-impeller design gives a low efficiency and is used only in the cheaper pumps for handling thick liquids such as mash, pulp and sewage. The enclosed or shrouded type of impeller, Figs. 223 and 227, consists of blades and side plates forming enclosed water passages. The water enters the passages at the center and the enclosing walls prevent water from escaping past the impeller blades. Where an enclosed type impeller is employed a double suction is used thereby minimizing end-thrust on the shaft. Impellers are

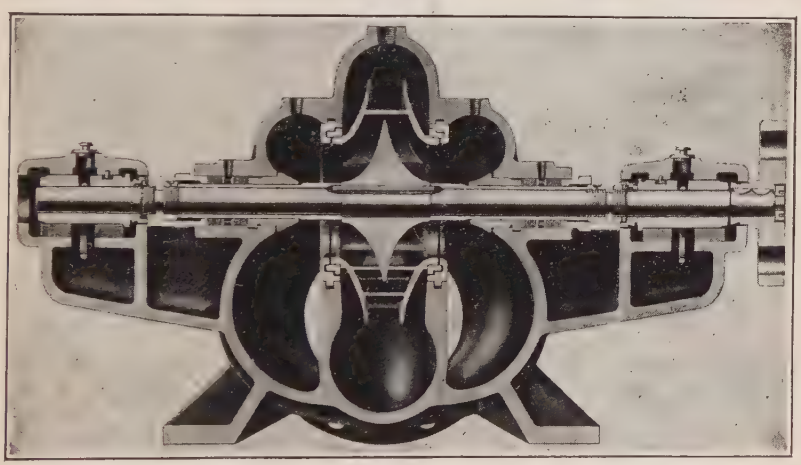


FIG. 226.—Axial section, De Laval single-stage, double-suction centrifugal pump.

usually made from bronze, which has proved best suited for withstanding the action of water and which retains its original shape and smoothness.

A vertical-shaft centrifugal pump may be necessary where conditions are such that a horizontal-shaft pump cannot be placed within suction distance (less than 23 ft.) of the supply water level. This condition is frequently encountered in deep wells, sewage service, sumps and near rivers where the difference in water level varies considerably. The pump may be completely submerged in the water but when possible is located in a dry pit, making the pump easily accessible. The bearings of vertical pumps require considerable attention and hence the horizontal shaft type is used wherever feasible.

256. Comparison of Pumps.—Piston pumps have a wide field of application in small power plants and for certain industrial purposes,

especially where the quantity of fluid to be handled is small and the total pressure pumped against is high. The advantages of a piston pump are that the action is positive, the pump does not " lose its water " and will generally operate when in a very indifferent state of repair. The pump input or power required can readily be determined; the speed alone governs the quantity of water pumped. Where either the head or the discharge (quantity) are required to vary within wide limits and where

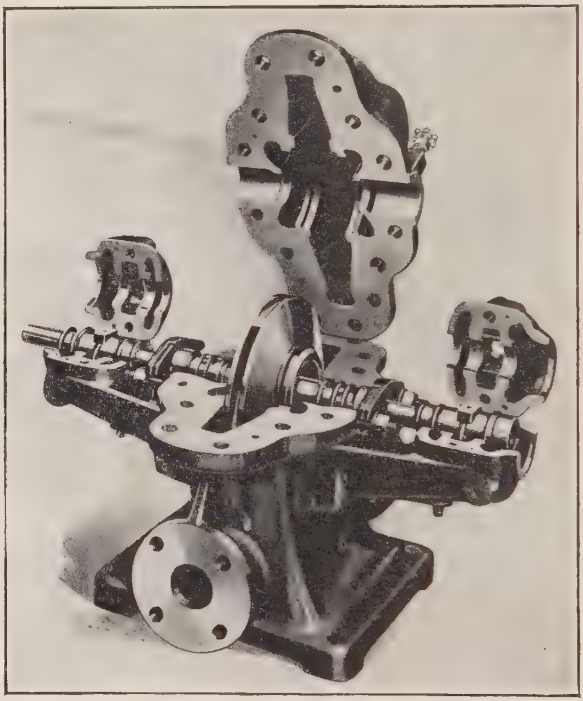


Fig. 227.—Single-stage double-suction centrifugal pump.

they do not maintain definite relations to each other, the displacement pump will be found to be flexible and economical. Direct-acting steam pumps are very wasteful of fuel and are low in thermal efficiency, largely because of the non-expansive use of steam and slow-speed operation. The simple or simplex pump generally uses less steam for the same amount of work than does the duplex pump on account of smaller clearance spaces in the steam cylinder of the former type. The piston of the simple pump travels the maximum set distance during each stroke, whereas the pistons of the duplex pump may " short-stroke " due to the retarding effect of friction between the pistons and cylinder walls and

friction in the piston-rod stuffing boxes. Simple pumps are well adapted for vacuum pumps in connection with condensers as the steam cylinder clearances are comparatively small and there is no short-stroking. The practically continuous action of duplex pumps makes them better adapted for high-pressure service than simple pumps; the former are also preferable where either a very high or a very low velocity of flow is required.

Flywheel and power pumps as compared with direct-acting steam pumps have a higher overall efficiency, may be run at higher speeds, there is no danger of short-stroking, and less clearance is necessary. The first cost of direct-acting pumps is less, also the attendance and maintenance required are less than for flywheel and power pumps. However, power pumps may be located many miles from the source of power and automatic control may be effected readily when electrically driven.

The lack of balance of piston and plunger pumps necessitates costly foundations. The repeated reversals of flow of the fluid cause stresses which may shatter the pump or its connections. These pumps are bulky and costly for a given output, and often the extra interest on the first cost and maintenance offsets any benefits from a probable higher efficiency. Without a flywheel, piston pumps are difficult to drive by belt, gas engine or electric motor. Piston and plunger pumps cannot satisfactorily handle dirty, gritty and highly viscous fluids.

The centrifugal pump with steam-turbine or electric-motor drive combines maximum efficiency in power transmission with simplicity, compactness, balanced rotary motion, absence of valves and pistons, uniform discharge, reliability and small weight. For pumping small volumes of water against high pressures, the centrifugal pump can be connected directly to, and run at the same speed as, the turbine. Where the proper speeds of the pump and turbine are too far apart to permit satisfactory efficiency at a compromise speed, or where the quantity of water to be handled is very large, speed-reducing gears are interposed between the turbine and pump. Boiler-feed pumps, condensate pumps, hydraulic-pressure pumps, elevator pumps, fire pumps, mine pumps, and other similar-purpose pumps are usually directly connected to the turbine. Condenser-circulating pumps, large general water-service pumps, and water-works pumps are generally driven by gears. The centrifugal pump discharges a continuous stream of fluid. Originally developed as a low-lift pump, the centrifugal pump is now used for high-pressure service and pressures exceeding 850 lb. per sq. in. can be obtained in an 8-stage unit. This type of pump is less costly to install, operate and maintain than the piston type, and will pump water con-

taining mud and small stones. Some of the disadvantages of centrifugal pumps are: the efficiency is not usually as high as the best grade of piston pump, and the pump is not satisfactory where a high-suction lift is necessary or for pumping small quantities of water against high-discharge pressures. A centrifugal pump when designed for a certain capacity and total pressure cannot be used without considerable loss of efficiency at any other capacity or pressure. The reciprocating pump is more flexible in this respect, as it can be operated under widely different conditions without any large sacrifice of efficiency.

257. Performance of Pumps.—Suitability of a pump for a given service is based upon the relations of head, capacity, speed, power and efficiency. The relation between the fundamental variables, head and capacity, may be represented by means of a curve; the curves for efficiency and brake horsepower may also be plotted. These curves are based on actual test results and vary with the design of the pump, and when taken together form the " characteristics " of the pump. Other curves may be obtained, however, by keeping any one of the fundamental quantities constant and varying the others.

The total *head* againt which a pump is delivering water is equal to the sum of the change of velocity head in the pump and the difference between the head with which the water leaves the pump and that with which the water enters the pump.

$$h_t = h_d - h_s + \frac{V_d^2 - V_s^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad (144)$$

where h_t = total head in feet of water.

h_d = discharge head in feet of water, as measured by a gage at the center of the pump.

h_s = total-suction head in feet of water, including velocity head in suction pipe, as measured by a gage, the connection being at the center of the pump. When water is lifted by suction h_s is considered negative, and when water flows under pressure to the pump h_s is positive. See Example.

V_d = velocity of water in discharge pipe, ft. per sec.

V_s = velocity of water in suction pipe, ft. per sec.

g = acceleration due to gravity, 32.17 ft. per sec.^{2}

In the above equation the small losses in the pump valves are neglected.

Suction lift is the vertical distance from the level of the water to be pumped to the center line of the pump. When the pump lifts the water

from a level below the center line, the pressure in the suction line is less than atmospheric and h_s is considered negative. If this water level is above the center of the pump, the pump then receives water under pressure and h_s is considered positive. Discharge head is the vertical distance between the center line of the pump and the level (or its equivalent) to which the water is elevated. The friction head may be determined by the use of charts or tables to be found in any standard handbook or textbook on "Hydraulics," which give the frictional losses in pipes, fittings and valves. These frictional losses are necessarily approximate and depend on such factors as smoothness of pipe, fluid handled, length of time in service, type of valves, radius of bends, etc. The velocity head, sometimes known as the head due to velocity, is the equivalent head through which the water would have to fall in order to acquire the velocity at a given point, that is, $h = \frac{V^2}{2g}$. The change of velocity head $\frac{V_d^2 - V_s^2}{2g}$ should be considered, however, when the discharge pipe is of different size (generally smaller) than the suction pipe, when the total head is low and when the suction lift is high, or in accurate testing.

Example.—The diameter of the discharge pipe of a boiler-feed pump is 4 in. and the suction pipe is 6 in. The water delivered is 250 gal. per minute and the temperature is 60 deg. fahr. The discharge pressure near the pump is 190 lb. per sq. in. gage and the suction is 12 inches of mercury vacuum. Find the total head pumped against if the discharge gage is located 3 ft. above the center of the pump and the vacuum-gage *connection* to the suction pipe is 2 ft. below the center of the pump. Assume that the head necessary to overcome the frictional resistance through the pump is 2.20 ft.

Solution.—Weight of water at 60 deg. fahr. = 62.37 lb. per cu. ft., from Table XIV (end of book).

$$h_d = \frac{190 \times 144}{62.37} + 3 = 441.7 \text{ ft., discharge head.}$$

$$h_s = \frac{-12 \times 13.59}{12} - 2 = -15.59 \text{ ft., suction head.}$$

$$\text{Quantity flowing} = \frac{250 \times 231}{1728 \times 60} = 0.557 \text{ cu. ft. per sec.}$$

from the equation, $Q = AV$, that is, quantity = area $\times$ velocity.

Hence, $$V_d = \frac{\text{cubic feet per second}}{\text{area of 4-in. pipe}} = \frac{0.557}{0.0873} = 6.38 \text{ ft. per sec.}$$

$$V_s = \frac{\text{cubic feet per second}}{\text{area of 6-in. pipe}} = \frac{0.557}{0.1963} = 2.84 \text{ ft. per sec.}$$

NOTE: Areas from Table XV.

Substituting in (144)

$$h_t = 441.7 - (-15.59) + 2.2 + \frac{(6.38)^2 - (2.84)^2}{2 \times 32.17}$$

and $$h_t = 460 \text{ ft. of water.}$$

The *capacity* of a pump is generally stated as the quantity of water the pump will discharge, expressed in gallons per minute. For piston pumps, the capacity is usually given as the piston displacement in cubic feet per minute. For a centrifugal pump the quantity varies directly as the speed, the head as the speed squared, and the power as the speed cubed. *Pump slip* is the difference between the displacement or theoretical discharge of a pump and the actual discharge, and is commonly expressed as a percentage of the displacement. Average values of pump slip range from 3 to 25 per cent. The *volumetric efficiency* of a piston pump is the ratio of the volume of water actually delivered and the displacement of the pump.

The hydraulic or *water horsepower* is the power actually delivered to the water, as computed from the weight of water actually discharged and the total head.

$$\text{W. hp.} = \frac{W_w \times h_t}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad (145)$$

in which W_w = pounds of water delivered per minute.
h_t = total head in feet of water.

Mechanical efficiency is the ratio between the power delivered to the water and the power supplied to the pump. Or

$$e_m = \frac{W.\ hp.}{Brake\ hp.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (146)$$

The brake horsepower is considered as the power delivered to the pump shaft by motor, turbine or pulley.

The *duty* of a steam-driven pump is the number of foot-pounds of work done by the water piston per million B.t.u. consumed by the steam end. The term duty may also be applied to a motor-driven pump if

the work done per million B.t.u. supplied to the motor is determined. Duties in excess of 100,000,000 ft.-lb. have been obtained with turbine-driven centrifugal pumps.

$$\text{Duty} = \frac{60W_w \times h_t}{W_s(H_1 - h_2)} \times 1{,}000{,}000 \quad . \quad . \quad . \quad . \quad (147)$$

W_w and h_t have been defined with equation 145.
W_s = weight of steam supplied, lb. per hr.
H_1 = total heat per lb. of steam supplied, B.t.u. above 32 deg. fahr.
h_2 = heat of the liquid per pound of steam at exhaust pressure, B.t.u. above 32 deg. fahr.

Example.—A 10-in. × 6-in. × 10-in. direct-acting duplex pump with $1\frac{1}{2}$-in. piston rods makes 70 single strokes per minute and delivers 135 g.p.m. of 100 deg. fahr. water against a total head of 460 ft. The steam is supplied to the steam end at a pressure of 150 lb. per sq. in. gage and exhausts to the atmosphere. Amount of steam supplied is 3800 lb. per hr. at 98 per cent quality. Barometer, 29.6 in. mercury. Find (*a*) the piston displacement of the pump in cubic feet per minute, (*b*) pump slip, (*c*) water horsepower delivered, and (*d*) the duty of the pump.

Solution.—

$$(a)\ \textit{Piston displacement} = \frac{.7854\,[36 + (36 - 2.25)] \times 10}{1728} \times 70$$

$$= 22.15 \text{ cu. ft. per min.}$$

$$(b)\ \textit{Volumetric efficiency} = \frac{135 \times 231}{1728} \times \frac{1}{22.15} \times 100 = 81.5 \text{ per cent.}$$

$$\textit{Pump slip} = 100 - 81.5 = 18.5 \text{ per cent.}$$

$$(c)\ \textit{Water horsepower} = \frac{460 \times 135 \times 231 \times 62}{1728 \times 33{,}000} = 15.6$$

$$(d)\ \textit{Duty} = \frac{460 \times 135 \times 231 \times 62 \times 60}{1728 \times 3800[337.5 + (0.98 \times 858.5) - 179.4]} \times 10^6$$

$$= 8{,}150{,}000 \text{ ft.-lb. per million B.t.u. consumed.}$$

The characteristic curves, Fig. 228, for pumps are usually based upon a constant speed of the pump, the curves showing the relation between head, capacity, horsepower and efficiency at that speed. The highest point on the total head curve is 51.5 ft., hence the pump could not be used for a greater head than this, as it would deliver no water, speed

remaining constant. The efficiency curve has a very gradual curvature which is a desirable characteristic, as it shows that the pump can be operated over a wide range of capacity with nearly uniform efficiency. The brake horsepower is a maximum at a point nearly coincident with the peak of the efficiency curve. The power curve and the efficiency curve

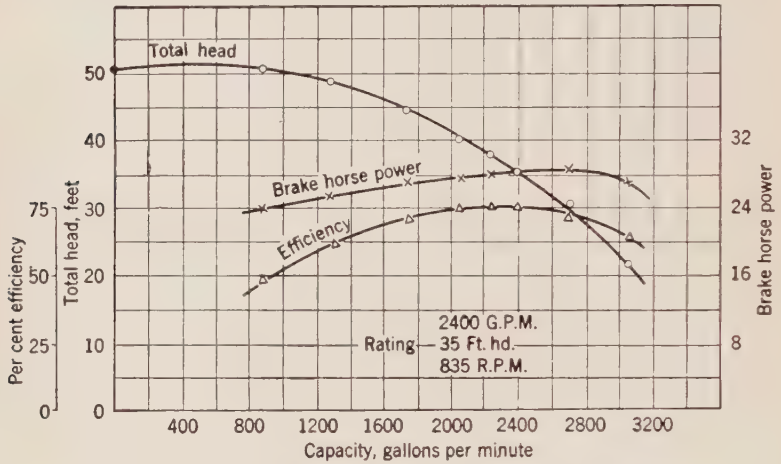


Fig. 228.—Characteristics of a 10-in. Wheeler centrifugal pump.

are usually a maximum at a capacity for which the pump was designed and then decrease with a further increase of capacity. The brake horsepower decreases for higher rates of discharge, thus the maximum capacity of the driving unit required is but little more than the power necessary under normal operation.

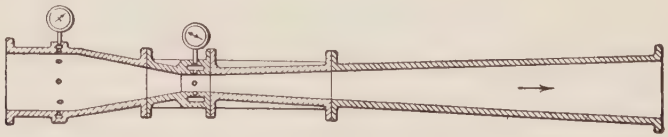


Fig. 229.—Section through a Venturi tube.

258. Measurement of Water.—Methods of water measurement are divided into primary and secondary, according to whether the method is an absolute measurement of quantity or merely the measurement of an *effect* of quantity or rate of flow. Of these methods the only primary methods are those of measurement by volume or weight. Secondary methods are more generally used in practice, and include measurement by Venturi tube, weir, orifice, nozzle, or pitot tube.

For the measurement of flow in pipes under pressure the *Venturi tube*, Fig. 229, is a reliable form of meter and is extensively used where accurate and consistent results are desired. The tube takes the shape of two truncated cones, joined at their smallest diameters by a short throat section, and inserted between flanges in a pipe line. Suitable pressure connections are provided for observing the pressure at the entrance and at the throat of the Venturi tube. The entrance cone has a total angle of about 21 deg. and leads to the short cylindrical throat section which is accurately machined. The diameter of the throat is usually between one-fourth and one-half of the pipe-line diameter. The exit end of the throat leads to the exit cone or diffuser which has a total angle of 6 deg. The maximum discharge through a Venturi tube is obtained when the throat pressure is 53 per cent of the entrance pressure.

The equation [1] for determining the quantity of water flowing through a Venturi tube is

$$Q = \frac{CA_1\sqrt{2gh}}{\left[\left(\frac{D_1}{D_2}\right)^4 - 1\right]^{\frac{1}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (148)$$

in which Q = rate of discharge, cu. ft. per sec.
C = coefficient, usually 0.96 to 0.99.
A_1 = area of pipe at upstream pressure connection, sq. ft.
g = acceleration due to gravity, 32.17 ft. per sec.^{2}
h = difference in head, corresponding to the difference between the static pressures at the two "piezometer" connections, ft. of water.
D_1 = diameter of pipe, upstream.
D_2 = diameter of pipe, throat.

The coefficient C varies with the rate of flow, diameter of the pipe, smoothness of inside surface, viscosity of the fluid, and other factors. In practice the numerical value of C is determined from experimental tests.

Example.—The inlet diameter of a Venturi tube is 6 in. and the throat diameter is 2 in. Water flowing through the tube has a temperature of 150 deg. fahr., an inlet static pressure of 160 lb. per sq. in. and throat static pressure 96 lb. per sq. in. How many gallons per minute is the pump delivering? Assume coefficient as 0.99.

Solution.—

$$h = \frac{(160 - 96)144}{61.2} = 150.6 \text{ ft. of water}$$

[1] For derivation see "Treatise on Hydraulics," p. 89, Tenth Edition, 1916, Mansfield Merriman, John Wiley and Sons, Inc.

From equation (148), $Q = \dfrac{0.99 \times 0.1963 \sqrt{2 \times 32.17 \times 150.6}}{[(\frac{6}{7})^4 - 1]^{\frac{1}{2}}}$

$$Q = \frac{19.13}{8.95} = 2.14 \text{ cu. ft. per sec.}$$

or

$$Q = \frac{2.14 \times 60 \times 1728}{231} = 961 \text{ g.p.m.}$$

The *weir* is an accurate means of determining the rate of flow of large quantities of water in open conduits. There are four principal types: the square notch; the rectangular weir; the V-notch, Fig. 230, which increases the head readings at low rates of flow; and the parabolic notches, which are so shaped as to give head readings directly proportional to the flow. The rectangular weir is generally used for high heads, such as may be found in the measurement of large streams. When the heads are relatively low (less than 12 in.), as in most power-plant service, the V-notch is more reliable. In the weir the variable area of the stream bears a definite relation to the velocity and is indicated by the height of the level of the measured liquid over a horizontal plane taken as a base. This variable height is known as the head h, of the weir and is the difference of elevation in feet between the surface of the liquid in the channel at a point placed just ahead of the beginning of the surface curve and the bottom or sill of the notch. The head of water passing over the V-notch, Fig. 230, is measured by a hook gage which is located at a point where flow occurs without turbulence and ahead of the beginning of the surface curve. The turbulence of the water flowing through a weir tank may be minimized by properly locating suitable baffles; the combing rack, Fig. 230, serves as an efficient means of straightening the

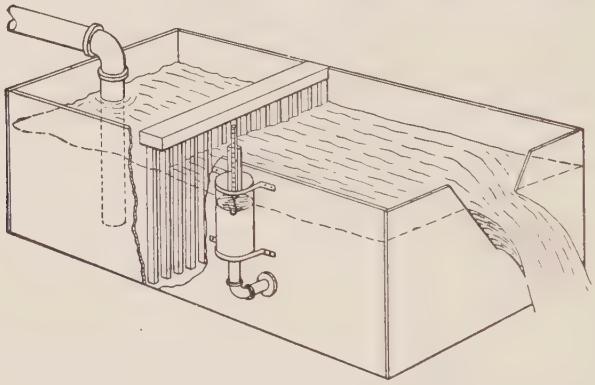


Fig. 230.—V-notch weir with combing rack and hook gage.

flow. For the rectangular-notch weir[1] with two contractions and heads ranging from 3 in. to 2 ft., experiments have shown that

$$Q = \tfrac{1}{3}^{0}(b - 0.2h)h^{3/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (149)$$

where Q = quantity of flow in cubic feet per second.
b = base of the rectangular notch in feet.
h = head of liquid passing over the weir in feet.

For the triangular weir or V-notch,[1] apex down, with angle at the apex of 90 deg., the rate of flow is

$$Q = 2.53h^{5/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (150)$$

PROBLEMS

1. During a pump test the pressure gage connected to the discharge pipe read 75 lb. per sq. in. and the suction gage recorded a vacuum of 5 in. of mercury. The center of the discharge gage was 2 ft. below the center of the pump shaft, while the vacuum-gage connection to the suction pipe was 4 ft. below the center of the shaft. The suction and discharge pipes were of the same size. Temperature of water, 120 deg. fahr. Find the total head in feet of water, neglecting friction between the gages. See Table XIV for density of water.

2. In a pump test, a mercury manometer used for measuring the discharge pressure reads 10 ft. of mercury. The suction gage recorded a positive pressure of 1 lb. per sq. in. The top of the lower column of mercury in the manometer was 2 ft. below the center of the pump, when above reading was observed; and the suction gage was 1 ft. above the center of the shaft. The diameter of the suction pipe was 5 in. and that of the discharge pipe 4 in. If the rate of discharge was 75 gallons per minute and the water temperature 200 deg. fahr., find the head developed. Add 2 per cent to the calculated head for frictional losses between gages. See Table XIV for density of water.

3. The water-piston diameter of a direct-acting simplex steam pump is 4 in. The stroke is 6 in. When the piston makes 80 strokes per minute, the volumetric efficiency is 95 per cent. Find the quantity of water discharged in gallons per minute.

4. A $4\frac{1}{2}$-in. $\times$ $2\frac{3}{4}$-in. $\times$ 4-in. duplex piston pump makes 100 single strokes per minute. The pump delivers 16 g.p.m., find the volumetric efficiency.

5. At 1800 r.p.m. a centrifugal pump delivers 2200 g.p.m. against a head of 120 ft. What will be the capacity and head at a speed of 1200 r.p.m.? What will be the speed and discharge for a head of 90 ft.?

6. A power pump delivers 100 g.p.m. of 50 deg. fahr. water against a total pressure of 250 lb. per sq. in. Find the water horsepower.

7. The diameter of the discharge pipe of a centrifugal pump is 6 in. and suction pipe is 8 in. Pressure gage reads 32 lb. per sq. in., vacuum gage reads 14 in. of mercury. If 3 cu. ft. of water at 90 deg. fahr. are delivered per second and the brake horsepower is 38, find the mechanical efficiency of the pump.

[1] "Treatise on Hydraulics," p. 154 (rectangular notch) and p. 169 (V-notch), Tenth Edition, 1916, Mansfield Merriman, John Wiley and Sons, Inc.

8. A compound duplex pump used 80 lb. of dry steam per i.hp.-hr.; i.hp. of steam end, 50; capacity 405 g.p.m. Temperature of water, 200 deg. fahr.; total head pumped against 180 lb. per sq. in.; steam supply pressure 120 lb. per sq. in. gage; moisture in steam, 3 per cent; exhaust pressure 22 in. mercury vacuum; and barometer 29.42 in. Find the duty per million B.t.u. consumed.

9. A pump delivers 800 g.p.m. against a discharge pressure of 50 lb. per sq. in. with a suction pressure of 10 in. of mercury, gage readings referred to the center of the pump. The efficiency (mechanical) of the pump is 62 per cent. What will be the horsepower required to drive the pump? Water temperature, 55 deg. fahr.

10. A pumping engine has a duty of 25,000,000 ft.-lb. per million B.t.u. when discharging 360,000 gal. of water per 24 hr. Steam pressure, 170 lb. per sq. in. (abs.) and 100 deg. fahr. superheat; exhaust pressure, 6 in. Hg (abs.). Discharge pressure of pump, 120 lb. per sq. in.; vacuum gage connected to suction pipe reads 10 in. Hg and the distance between center of pressure gage and point of attachment of vacuum gage is 7 ft. Mechanical efficiency of pump is 90 per cent. Determine the steam consumption per indicated horsepower per hour.

11. A 10-in. $\times$ 6-in. $\times$ 12-in. direct-acting duplex boiler-feed pump with a $1\frac{1}{2}$-in. rod makes 20 complete strokes per minute per cylinder and receives water in the suction pipe at a pressure of 5 lb. per sq. in. (by gage at center of pump), and at a temperature of 150 deg. fahr. The water is pumped through an economizer before entering the boiler where it has a temperature of 200 deg. fahr. and a pressure of 150 lb. gage (barometer = 29.42 in. Hg). The pressure on the discharge side of the pump (at center) is 175 lb. per sq. in.

(*a*) If the volumetric efficiency is 85 per cent, how many pounds of water will the pump deliver per hour? Use weight of water at suction temperature, Table XIV.

(*b*) How many boiler horsepower will the pump supply?

(*c*) Find the water horsepower delivered.

(*d*) If the mechanical efficiency is 70 per cent and the steam supplied to the pump from the above boiler is 200 lb. per i.hp. per hour and at 98 per cent quality, find the duty per million B.t.u. consumed, if the engine runs non-condensing.

CHAPTER XVIII

COMPRESSED AIR

259. Introductory.—When large volumes of air are delivered at low pressures, less than 10 in. of water, fans are generally employed (see Chapter X); the air may be used for ventilation, combustion of fuel in boiler furnaces and similar purposes. Air pressures up to 10 lb. per sq. in. gage are produced by rotary blowers, as used for supplying air for cupola furnaces, or air to oil and gas burners. The centrifugal compressor or turbo-compressor, which is somewhat similar to a centrifugal pump, may be used for air pressures up to 20 lb. per sq. in. gage with a single-stage unit and for higher pressures, up to 125 lb. per sq. in. gage, with the multi-stage types. Centrifugal compressors may be used in furnishing an air blast for smelting furnaces, Bessemer converters, gas producers and in "boosting" gas for high-pressure distribution. The reciprocating piston compressor is usually employed for producing higher pressures. If the final air pressure is to exceed 100 lb. per sq. in. gage, two or more stages of piston compression are usually employed.

Where power is required intermittently at a distance from its source, compressed air is particularly valuable because the air pressure can be maintained nearly constant during intervals of work. Compressed air may replace steam when power is to be transmitted over a considerable distance, because the steam condensation loss would be excessive, as exemplified in the use of compressed air for mining, air-brake service, sand blasting and tunneling operations.

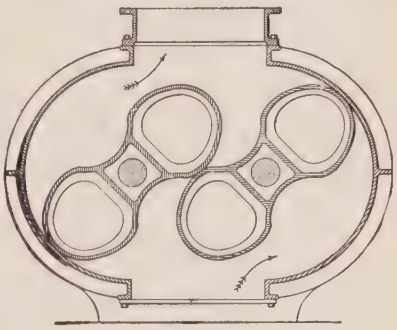


Fig. 231.—Interior view of Roots Acme blower.

260. Rotary Compressors.—For pressures from $\frac{1}{2}$ to 10 lb. per sq. in. gage, *rotary blowers* such as that shown in Fig. 231 are used. Two impellers mounted on parallel shafts rotate in opposite directions. There is no actual contact between the impellers and the casing, and the impellers

themselves. There are no valves or restricted passages through which the air must pass. A rotary blower of this type having 4-in. inlet and discharge openings, and operating at 550 r.p.m. can deliver 274 cu. ft. of air per minute against a pressure of 2 lb. per sq. in. gage. Rotary blowers are employed to furnish air to oil and gas burners, agitation processes, air brushes, spraying equipment, torches, cleaning nozzles and forges. They may also be used to produce a partial vacuum up to 16 in. of mercury on the suction side of the blower.

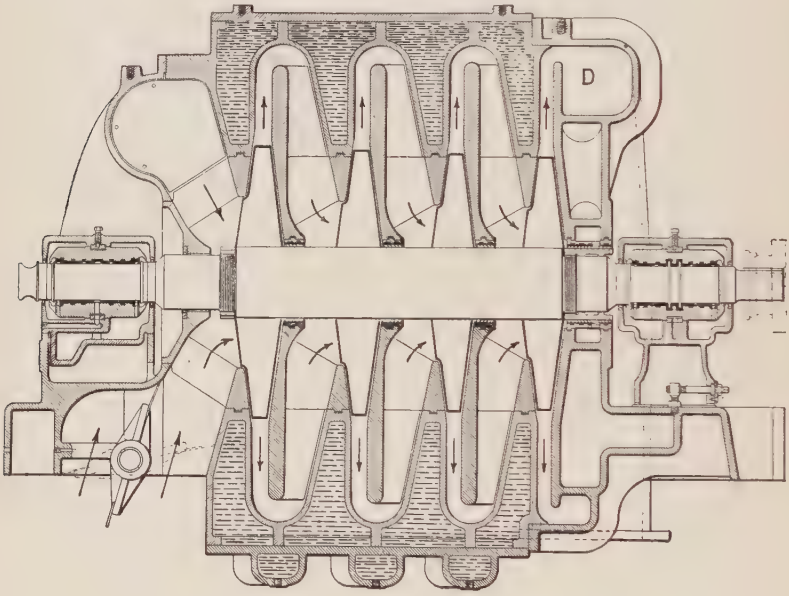


Fig. 232.—Section of Elliott four-stage blower.

Centrifugal compressors, also called turbo-compressors, are used for air pressures of 10 to 20 lb. per sq. in. gage with a single-stage unit. For pressures up to 125 lb. per sq. in. gage the compression is done in several stages; the pressure obtained depends primarily upon the speed and the number of stages or impellers used. Fig. 232 shows a four-stage centrifugal compressor. The rotor, Fig. 233, consists of straight radial impellers made of high tensile-strength alloy steel; this construction requires less weight of material than the shrouded type, see Fig. 227. The rapid rotation of the impellers (rarely less than 3000 r.p.m.) imparts to the air a velocity of 300 ft. per sec. or more. The velocity at which the air issues from each impeller is converted into pressure as the air passes into the larger passages leading to the next impeller, each adding

an increment of pressure. Between the impellers are mounted curved guide vanes which direct the flow of air discharged by one impeller into the next stage at the proper angle in the direction of rotation. In multi-stage casings, water cooling is usually provided in the spaces between the impeller casings and passages, Fig. 232, for the partial removal of the heat of compression. A three-stage compressor of this type operating at 3600 r.p.m. can deliver 60,000 cu. ft. of air per minute against a pressure of 30 lb. per sq. in gage. Centrifugal compressors are used for supplying compressed air for metallurgical service, exhausters, evaporators, grain elevators, pneumatic transmission, tools, sewage-disposal plants, sand blast, and gas boosters.

261. Piston Air Compressors.—The construction of the air cylinder of a piston compressor is, in general, similar to that of the cylinder in a

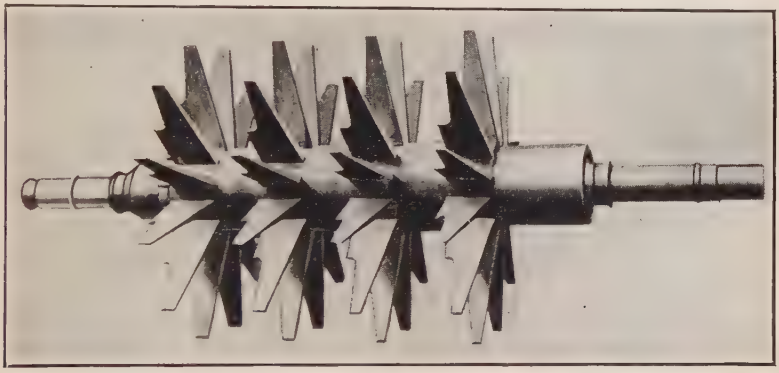


FIG. 233.—Rotor of Elliott four-stage blower.

steam engine. In a double-acting compressor each end of the cylinder must be fitted with an inlet and discharge valve corresponding to the inlet and exhaust valves of the steam cylinder. The valves may be actuated mechanically or by the air itself. The driving power for an air compressor is either a steam engine, electric motor, internal-combustion engine, or water wheel. The essentials of a piston air compressor are shown in Fig. 234, I being a spring-closed inlet valve and D the discharge valve. The indicator (or PV) diagram in Fig. 234 shows the changes of state of the air in its passage through the compressor.

Single-Stage Compressor.—Air, Fig. 234, is drawn into the cylinder from the atmosphere along the line AB. The pressure in the cylinder during the suction stroke is below that of the external or atmospheric pressure P_1. This can be accounted for by the fact that the pressure in the cylinder must be lower than that outside before the inlet valve will open. At B the piston starts on its return stroke and the inlet valve

closes. If there were no heat loss due to radiation and no friction losses, the compression of the air would be adiabatic and would follow the line BC'. At point C' the discharge valve would open and the air would be forced into the receiver, the pressure following the line $C'D$. At D the discharge valve closes and the air in the clearance space expands from D to A as the piston starts on its return stroke. At A the inlet valve opens and the cycle starts again. If, by some means, the heat caused by compression could be taken away as fast as formed, the compression of the air would be isothermal and would follow the line BC''. The remainder of the cycle would be the same as before.

By reference to the PV diagram in Fig. 234 it can be seen that an equal weight of air can be compressed more economically by isothermal compression than by adiabatic compression. The area inside the figure $ABC'D$ represents the *work done on the air* during adiabatic compression. $ABC''D$ represents the work done on the air during isothermal compression, and area $C''BC'$ represents the saving effected by the use of isothermal instead of adiabatic compression. Efforts have been made to approach isothermal compression by: (1) injecting cold water as a spray into the air cylinder, (2) cooling the cylinder from without, that is, enveloping it in a cold water jacket, and (3) dividing the work into two or more steps or stages with an intercooler between successive stages. The use of either an atomized spray or a solid jet of cold water is effective as a means of cooling but introduces difficulties in the lubrication of cylinder and piston. In American practice the injection of water into the air cylinder has been abandoned, and it is generally customary to water-jacket piston air compressors. The air can be cooled partially with a water jacket and better cylinder lubrication obtained. The work done on the air by the *actual* single-stage compressor using a water jacket can be represented by the figure $ABCD$. It should be noted that the discharge line CD of this diagram lies above the receiver pressure line P_2. Discharge occurs along CD because the pressure inside the cylinder must be greater than that outside before the discharge poppet valve will open and allow the air to escape.

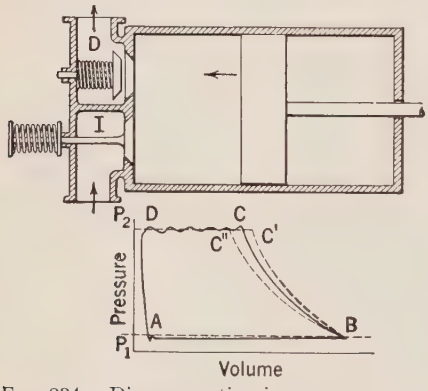


FIG. 234.—Diagrammatic air compressor and PV diagram

For the adiabatic compression curve BC', Fig. 234, $PV^k = C$. Where $k = 1.40$ and for the isothermal compression curve BC'', $PV = C$. Because of the inability of the water jacket to absorb heat rapidly, compression occurs along the curve BC and $PV^n = C$ where the exponent n lies somewhere between 1 and 1.40. Under usual working conditions, n is between 1.28 and 1.35.

Multi-stage Compressors.—An effective method of approaching isothermal air compression (low value of n) is by multi-stage operation, the principle of which is illustrated by Fig. 235. Assume that A is the state at the beginning of compression, and that the path AB is practically adiabatic, even though jacket cooling is used. Let AF be an isothermal drawn through A. In multi-stage compression, the air follows path AB to some intermediate pressure P' at C. The air is discharged and cooled at constant pressure in an intercooler (external, water cooled) until the temperature of the air is approximately the same as at admission to the cylinder. CD represents this cooling. The air now passes to the second cylinder, where it is adiabatically compressed along DE and discharged along EF at pressure P_2 to the receiver. The diagram illustrates two-stage compression, but three or more stages are sometimes used. The work saved by the above two-stage compression over that of single-stage adiabatic compression is shown by the shaded area $CDEB$. If the final pressure exceeds 100 lb. per sq. in., the compression is more economically carried out in two or more stages. Multi-stage compression reduces the total work of compression. The mechanical stresses in the compressor are smaller than in a single-stage compressor of equal capacity and the temperature range in each cylinder is less than in a single cylinder. Thermodynamically, the efficiency of stage compression depends largely on the proper use of water jackets for the air cylinders, and the size and design of the intercoolers.

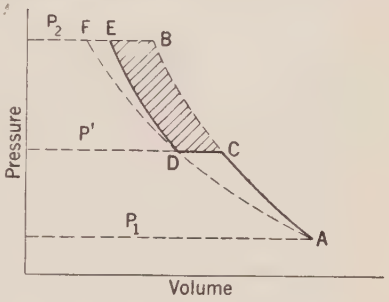


FIG. 235.—Multi-stage compression.

262. Compressor Classification and Details.—Reciprocating air compressors may be classified according to the method of driving, number of stages, type of frame, method of cooling, kind of valves, etc. Steam-driven compressors are generally mounted as one unit with the steam cylinders and a common flywheel. Regulation is usually effected by controlling the speed of the steam engine in accordance with the air

pressure in the receiver, however, the steam-engine governor imposes a maximum speed limit. Compressors are also operated by electric motors, belt drives, internal-combustion engines or water wheels. Constant-speed compressors may be regulated by means of an " unloading " valve, which either keeps the suction valve closed during one or more strokes or allows the air to discharge into the atmosphere, when the desired receiver pressure is reached. One prominent manufacturer regulates the constant-speed compressor by a patented 5-step clearance volume control which loads or unloads the compressor in five successive steps, viz., full load, three-quarters, one-half, one-quarter and no load. With this clearance control a number of clearance pockets in the high- and low-pressure cylinders are thrown into communication with each cylinder during compression in proper sequence. the process being con-

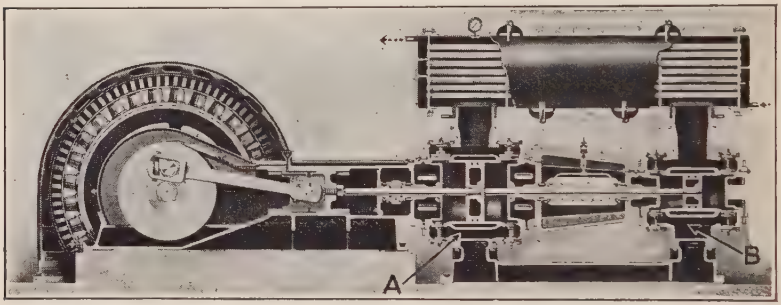


Fig. 236.—"Chicago Pneumatic" tandem air compressor.

trolled by a predetermined air-receiver pressure. High-compression efficiencies are obtained at all loads and the power required is in proportion to the load. In the variable-discharge type of compressor, the delivery may be regulated by closing the inlet valve before the completion of the forward (suction) stroke.

The ordinary small compressor is a single-stage reciprocating machine, with poppet air valves in the head or sides of the cylinder. Different arrangements of cylinders and pistons are possible in multi-stage compressors, the most common being the tandem and the cross-compound. Fig. 236 shows a motor-driven two-stage double-acting compressor with the low-pressure and high-pressure cylinders in tandem with an intercooler between the stages. The inlet pipe connection to suction space A is not shown by Fig. 236. The pipe connection to discharge space B of the high-pressure cylinder has also been cut away. The intercooler consists of a cast-iron cylindrical shell, inside of which are straight brass tubes between the tube sheets. The air entering the shell

from the first stage surrounds the tubes and is compelled by baffles to cross the tube space several times on the way to the second stage. The cooling water is guided to the tubes by projecting ribs on the intercooler heads, which cause the water to pass from end to end several times. The design of the intercooler should be such as to insure efficient heat transfer without undue pressure loss. The pressure loss through the intercooler may be caused by shrinkage of air due to cooling or frictional resistance to the flow of air. The air valves in Fig. 236 are of the thin-plate spring-seated type, similar to Fig. 238. The tandem compressor is largely employed for small or medium-sized plants; the machine

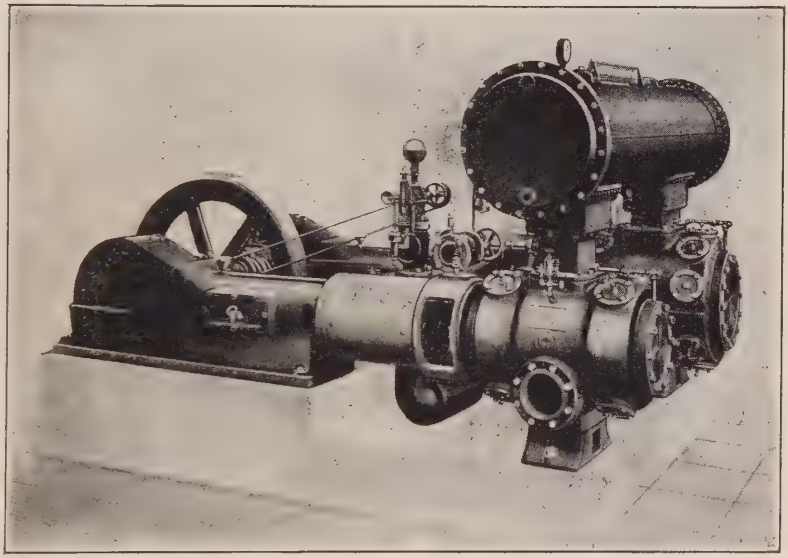


FIG. 237.—Cross-compound air compressor.

being compact, strong and self-contained. While useful for moderate air pressures and fairly constant loads, the steam-driven tandem-compressor is not capable of operating with good steam economy, nor is it self-regulating at less than 40 per cent of its rated load. These compressors are built up to capacities of 2000 cu. ft. of "free air" [1] (14.7 lb. per sq. in. and 60 deg. fahr.) per minute and 350 hp.

The cross-compound compressor, Fig. 237, is preferable to the tandem for continuous service in large plants. It is better adapted to varying loads, is self-regulating, gives good steam economy down to one-

[1] Cubic feet of free air is also understood to be the actual amount of air drawn into the cylinder, measured at the temperature and pressure of the atmosphere.

fourth rated load, and its frictional losses are generally lower than for the tandem type. The slide-valve steam engine is often used for driving small cross-compound compressors. The Corliss type of engine is frequently used for the larger sizes, as the valve gear is well adapted for dealing with variation of air pressure. Mechanically-actuated valves (Corliss or poppet) may be used for the air as well as the steam cylinders. Several well-known manufacturers have adopted the automatically-opened thin-plate (Fig. 238), "Feather" type, or other kinds of air valves. The cross-compound installation requires more floor space and the first cost is higher than the tandem compressor. Standard machines of the cross-compound type are built in capacities up to 8000 cu. ft. of "free air" per minute.

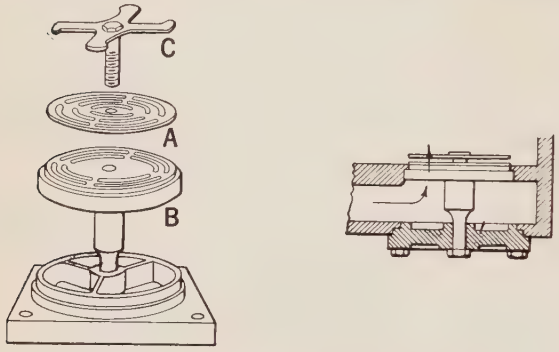


FIG. 238.—Plate-valve detail and assembly.

In most manufacturing plants, mines, quarries, etc., the presence of moisture in the compressed air is objectionable, on account of the difficulty of lubrication of air tools, liability of freezing and rapid destruction of air-hose connections. Aftercoolers are employed to remove the moisture (and oil) present after the last stage of compression and to cool the air, thus decreasing the necessary size of the transmission pipe. Aftercoolers are similar in design and appearance to intercoolers. Intercoolers and aftercoolers are provided with a drain-cock for removing the moisture that collects. The receiver tank is also provided with a drain-cock, and has a safety valve and pressure gage.

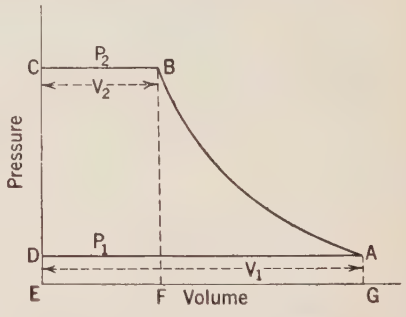


FIG. 239.—Theoretical PV diagram for air compressor without clearance.

263. The Ideal Reciprocating Air Compressor.—The area $ABCD$, Fig. 239, represents the net work delivered to the air by a compressor without clearance. Let P_1, V_1, and P_2, V_2, respectively denote the

pressures and volumes at points A and B. The net work in ft.-lb. done on the air per cycle is

$$W = W_{ab} + W_{bc} - W_{da}$$

where W_{ab} = work of compression along AB, area $GABF$.
W_{bc} = work of discharging the air from the cylinder, area $FBCDE$.
W_{da} = work done by atmospheric pressure on the piston during suction, area $DAGE$.

Assuming *adiabatic* compression, from equation 38, Chapter II,

$$W_{ab} = \frac{P_2V_2 - P_1V_1}{k - 1}$$

and

$$W = \frac{P_2V_2 - P_1V_1}{k - 1} + P_2V_2 - P_1V_1$$

or

$$W = \frac{k}{k - 1}(P_2V_2 - P_1V_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad (151)$$

but

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{k}}$$

Substituting and reducing,

$$W = \frac{k}{k - 1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] \quad . \quad . \quad . \quad . \quad (152)$$

which is the general expression for the work of single-stage compressors with adiabatic compression, when the clearance is zero.

For *polytropic compression* $PV^n = C$, from equation 37, Chapter II,

$$W_{ab} = \frac{P_2V_2 - P_1V_1}{n - 1}$$

therefore,

$$W = \frac{P_2V_2 - P_1V_1}{n - 1} + P_2V_2 - P_1V_1$$

or

$$W = \frac{n}{n - 1}(P_2V_2 - P_1V_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad (153)$$

but

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

Substituting and reducing,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad . \quad . \quad . \quad . \quad (154)$$

The foregoing is the general expression for work of single-stage compressors, when the compression is polytropic, according to the law PV^n = constant, when the clearance is zero.

For *isothermal* compression $PV = C$, from equation 41, Chapter II,

$$W_{ab} = P_1 V_1 \log_e \left(\frac{V_1}{V_2} \right)$$

Thus

$$W = P_1 V_1 \log_e \left(\frac{V_1}{V_2} \right) + P_2 V_2 - P_1 V_1$$

but

$$P_2 V_2 = P_1 V_1 \quad \text{and} \quad \frac{V_1}{V_2} = \frac{P_2}{P_1}$$

hence

$$W = P_1 V_1 \log_e \left(\frac{P_2}{P_1} \right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (155)$$

This is the general equation for the work of single-stage compressors operating isothermally, when the clearance is zero.

The volume of air at A, which has been designated by symbol V_1 in equations 151 to 155, is generally taken as " free air " at intake (point D) temperature and pressure. *Free air as commonly defined is air at* 14.7 *lb. per sq. in. absolute pressure and at a temperature of* 60 *deg. fahr.* Another interpretation of the expression free air is, free air is the normal atmospheric condition of the air at the point where the compressor is installed.

264. Net Work Done on the Air in Single-Stage Compression.—An actual compressor cannot be operated without cylinder clearance, as there must always be a certain amount of mechanical clearance between the cylinder head and the dead-center position of the piston to insure safe operation. The clearance volume is somewhat increased by the valve passages between the valve faces and the inside of the cylinder. The air remaining in the clearance space expands on the return stroke

of the piston, thus partially filling the cylinder and reducing the volume of external air taken in during the suction stroke. In Fig. 240, curve CD represents the "expansion" of the air in the clearance space. *The volume of the air entering the cylinder per stroke is represented by length* V_1, the piston displacement by V_p, and the clearance volume by V_c.

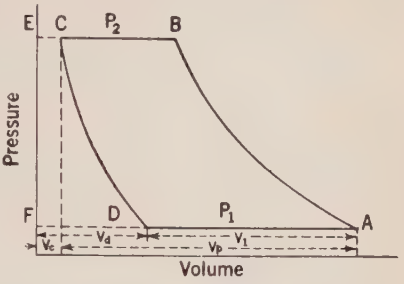


FIG. 240.—Theoretical PV diagram for air compressor with clearance.

If the value of n for the curve CD is the same as for the compression AB, the net work done on the air per cycle is given by the same expression as in the case of the compressor without clearance, equation 154,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

If curves AB and CD, Fig. 240, are adiabatic the work done on the air per cycle is equation 152,

$$W = \frac{k}{k-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]$$

and if isothermal by equation 155,

$$W = P_1 V_1 \log_e \left(\frac{P_2}{P_1} \right)$$

The clearance does not affect the net work delivered by the compressor; the air in the clearance space reacts like a spring, storing and giving back energy alternately. In other words, clearance affects the volumetric output of the compressor but not the work of compression for a given volume of air taken into the cylinder. The clearance of a compressor is the ratio of the clearance volume to the piston displacement, from Fig. 240, $\frac{V_c}{V_p} = m$, the clearance, and may vary from 1 to 5 per cent depending upon the size and design of the compressor.

The above work equations are sometimes more conveniently

expressed in terms of cylinder displacement and " volumetric efficiency " than in the pressure-volume units. Referring to Fig. 240,

V_p = displacement volume of cylinder, cu. ft.

$$m = \frac{V_c}{V_p} \quad \text{or} \quad V_c = mV_p$$

$V = V_p\,(1 + m)$, total cylinder volume in cu. ft.

V_d = volume of re-expanded clearance air in cu. ft.

Then

$$V_d = V_c\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} = mV_p\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

but

$$V_1 = V_p + V_c - V_d$$

Substituting,

$$V_1 = V_p + mV_p - mV_p\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

or

$$V_1 = V_p\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad (156)$$

The ratio of the free air actually taken into a compressor cylinder to the volume displaced by the piston is called the *volumetric efficiency* of the compressor, or $e_v = \frac{V_1}{V_p}$. Substituting the value of V_1 as obtained by equation 156, in equation 154,

$$W = \frac{n}{n-1}P_1V_p\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \quad . \quad . \quad (157)$$

which expresses the work in terms of displacement and clearance, with pressures in lb. per sq. ft. absolute.

But

$$e_vV_p = V_1 = V_p\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right],$$

Therefore

$$e_v = \left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$$

Substituting in equation 157, the expression for work done on the air reduces to

$$W = \frac{n}{n-1} e_v P_1 V_p \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad . \quad . \quad . \quad . \quad (158)$$

265. Work of Multi-stage Compression.—Figure 241 shows the theoretical PV diagram for two-stage compression with clearance. The curve AB represents the compression in the first cylinder to the intermediate pressure P', line BE the cooling of the air at constant pressure to the initial temperature T_1, curve EG the compression from P' to P_2 in the second cylinder. The diagram shows a single continuous polytropic expansion line CD, which is taken to represent the expansion lines of both cylinders. On the PV plane, $ABFD$ is the indicator diagram for the first compressor and $EGCF$ the indicator diagram for the second. For a single-stage compressor, working between the same pressure limits P_1 and P_2, the indicator diagram would be $AHCD$; hence the saving of work by two-stage compression over single-stage compression is represented by the area $BEGH$.

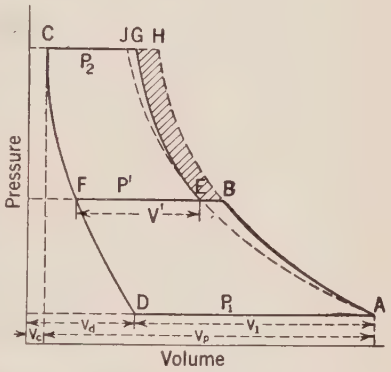


FIG. 241.—Work for two-stage compressor with clearance.

According to equation 154, the work done in the first cylinder is given by the expression,

$$W_1 = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P'}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

The work done in the second cylinder is given by the expression

$$W_2 = \frac{n}{n-1} P' V' \left[\left(\frac{P_2}{P'} \right)^{\frac{n-1}{n}} - 1 \right]$$

Since point E is on the isothermal AEJ, then $P'V' = P_1 V_1$ and therefore

$$W_2 = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P'} \right)^{\frac{n-1}{n}} - 1 \right]$$

Hence the general equation for the work of a two-stage compressor, with clearance and perfect intercooling takes the form

$$W = W_1 + W_2 = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P'}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_2}{P'} \right)^{\frac{n-1}{n}} - 2 \right] . \quad (159)$$

The pressure P_1 and P_2 are fixed, while P' is variable. By differentiating the above equation with respect to the variable P' and making the first differential coefficient $\frac{dW}{dP'} = 0$, it is found that the maximum value is given when

$$P' = \sqrt{P_1 P_2} \quad . . . . . (160)$$

or

$$\frac{P'}{P_1} = \frac{P_2}{P'} = \left(\frac{P_2}{P_1} \right)^{\frac{1}{2}} \quad . . . . . (161)$$

This is the value for the best (most economical) intercooler receiver pressure.

If there are three stages with intermediate pressures P' and P'' the condition for minimum work of compression is

$$\frac{P'}{P_1} = \frac{P''}{P'} = \frac{P_2}{P''} = \left(\frac{P_2}{P_1} \right)^{\frac{1}{3}} \quad . . . . . (162)$$

Combining equations 161 and 159, the total work of a two-stage compressor, with clearance and perfect intercooling and the intermediate pressure as given in (160) is,

$$W = \frac{2n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{2n}} - 1 \right] \quad . . (163)$$

Likewise for three stages,

$$W = \frac{3n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{3n}} - 1 \right] \quad . . (164)$$

Example.—Find the cylinder dimensions and horsepower required to drive an air compressor at 200 r.p.m. that is to compress 1000 cu. ft. of free air per minute to a pressure of 114 lb. per sq. in. abs. The exponent n for both compression and expansion will be assumed as 1.3 and the clearance ratio m as 0.03. Pressure at the beginning of compression is 14 lb. per sq. in. abs. and inlet temperature 60 deg. fahr. Piston speed to be 600 ft. per min.

Solution.—*Single-Stage Compressor.*

Air entering cylinder

$$V_1 = \frac{1000 \times 14.7}{14} = 1050 \text{ cu. ft. per min.}$$

Displacement volume, equation (156),

$$V_p' = \frac{1050}{1 + 0.03 - 0.03\left(\frac{114}{14}\right)^{\frac{1}{1.3}}} = 1193 \text{ cu. ft. per min.}$$

Displacement volume per cycle (double-acting),

$$V'_p = \frac{1193}{2 \times 200} = 2.98 \text{ cu. ft.}$$

Stroke, $$l = \frac{600}{2 \times 200} = 1.5 \text{ ft.} = 18 \text{ in.}$$

Effective piston area $= \frac{2.98}{1.5} \times 144 = 286.1$ sq. in.

Allowing 3.1 sq. in. for a 2-in. piston rod, the gross area should be 289.2 sq. in. and the corresponding diameter is $19\frac{1}{4}$ in.

The work per minute from equation (154) is

$$W = \frac{1.3}{0.3} \times 144 \times 14 \times 1050 \left[\left(\frac{114}{14}\right)^{\frac{0.3}{1.3}} - 1\right]$$
$$= 5{,}687{,}200 \text{ ft.-lb. per min.}$$

Therefore the power developed by the compressor $= \frac{5{,}687{,}200}{33{,}000} = 172.3$ air hp. If there is a 15 per cent loss between the steam engine and compressor the horsepower developed in the steam engine should be

$$172.3 \div 0.85 = 202.7 \text{ steam hp.}$$

Two-Stage Compressor

Intermediate pressure $P' = \sqrt{P_1 P_2}$

$$P' = \sqrt{14 \times 114} = 39.96 \text{ lb. per sq. in. abs.}$$

Assuming a pressure drop through the intercooler as 1 lb., take 40.5 lb. as the discharge pressure of the low-pressure cylinder and 39.5 lb. as the intake pressure to the high-pressure cylinder.

Low-pressure cylinder displacement volume,

$$V_p = \frac{1050}{1 + 0.03 - 0.03\left(\frac{40.5}{14}\right)^{\frac{1}{1.3}}} = 1092 \text{ cu. ft. per min.}$$

Using a stroke of 18 in., effective piston area

$$= \frac{1092}{2 \times 200 \times 1.5} \times 144 = 262 \text{ sq. in.}$$

Allowing 3 sq. in. for the piston rod, the corresponding low-pressure cylinder diameter is $18\frac{1}{2}$ in.

The air passing through the intercooler has its temperature reduced to the initial temperature of the air entering the low-pressure cylinder. Hence the ratio 14 to 39.5 is the relation between the air volumes of the high-pressure cylinder and the same volumes for the low-pressure cylinder.

High-pressure cylinder displacement volume $= 1092 \times \frac{14}{39.5} = 387$ cu. ft. per min.

Using a stroke of 18 in., effective piston area $= \frac{387}{2 \times 200 \times 1.5} \times 144$ $= 93$ sq. in.

Allowing 3 sq. in. for the piston rod, the corresponding high-pressure cylinder diameter is 11 in.

The work of the low-pressure cylinder is, from equation (154),

$$W_1 = \frac{1.3}{0.3} \times 144 \times 14 \times 1050 \left[\left(\frac{40.5}{14}\right)^{\frac{0.3}{1.3}} - 1\right]$$

$W_1 = 2{,}548{,}200$ ft-lb. per min.

Power developed in low-pressure cylinder $= \frac{2{,}548{,}200}{33{,}000} = 77.22$ air hp.

The work of the high-pressure cylinder is, from equation (154)

$$W_2 = \frac{1.3}{0.3} \times 144 \times 14 \times 1050 \left[\left(\frac{114}{39.5}\right)^{\frac{0.3}{1.3}} - 1\right]$$

$W_2 = 2{,}541{,}800$ ft-lb. per min.

Power developed in the high-pressure cylinder $= \frac{2{,}541{,}800}{33{,}000} = 77.02$ air hp.

Total air horsepower required for two-stage compression = 154.24, which is about 10 per cent less than was required for the single-stage compressor.

266. Compressor Indicator Diagrams.—The actual indicator diagram departs considerably from the ideal, the real compressor requiring a greater expenditure of work on the air than the ideal machine because

of clearance, leakage, friction, inefficient cooling, imperfect valve action, etc. Typical high- and low-pressure diagrams for two-stage compression are shown in Fig. 242. The indicator cards have been reduced to the same scale of volumes and pressures; the figure is called *the combined card*. The indicator card from each cylinder was divided into an equal number of parts and the pressures and volumes at each of these points were computed and plotted to obtain Fig. 242. Starting at D, the point of opening of the low-pressure suction valve, it should be noted that pressure P_1 is below atmospheric pressure. This is due to the fact that the internal pressure must be somewhat lower than the external pressure before the poppet valves will open or because of restricted inlet valve area. At A the suction valves close and the air is compressed along line AB. The adiabatic line AH falls to the right of AB and the saving due to water jacketing the cylinder is represented by area BAH. The discharge pressure in the low-pressure cylinder is greater than the intercooler pressure P', due to the friction and sticking of the discharge valves. Discharge in the low-pressure cylinder ceases at F and the clearance air expands from F to D before the inlet valve will again open. The wavy suction and discharge lines are caused by oscillations of the air column and oscillations or "fluttering" of the valves. AC is the isothermal compression line for the low-pressure cylinder. The area CAH represents the work that could have been saved if perfect cooling had taken place.

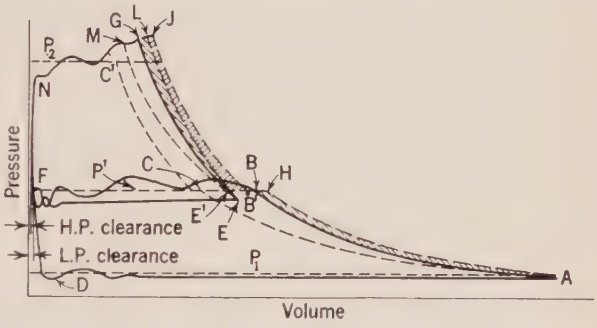


FIG. 242.—Combined card for two-stage compression.

If there were no difference in the clearance volumes of the two cylinders and the expansion lines crossed the intercooler pressure line at the same point, then, with no cooling in the intercooler, the compression in the high-pressure cylinder would be continued from point B, Fig. 242. Assuming no cooling in the intercooler the actual high-pressure cylinder compression would start at B' instead of B, due to the difference in clearance volumes in the two cylinders and adiabatic compression would take place along $B'J$. In the actual case the volume of air is decreased by cooling in the intercooler and the high-pressure compression starts at E instead of B'. Due to friction losses through ports and valves,

the actual suction line of the high-pressure cylinder is below the intercooler pressure P'. The actual compression line in the high-pressure cylinder is shown by line EG and the work saved due to high-pressure cylinder jacketing is shown by area GEL. Area $B'E'LJ$ represents the appreciable saving in work done due to intercooling. The isothermal compression line for the high-pressure cylinder is EM and the theoretical saving possible due to cooling in the high-pressure cylinder is represented by area MEL. The discharge valve does not open until the pressure attained in the high-pressure cylinder is above that of receiver pressure P_2. There are generally several oscillations of the valve and air column, and the discharge line generally obtained is somewhat like JMN. The discharge valves on some compressors do not close suddenly at the end of the stroke, consequently the corner at N may be more or less rounded.

The overlapping of the high- and low-pressure cards, Fig. 242, indicates a loss, because the work represented by this area is in reality work done twice. The drop in pressure between the cylinders is caused by the resistance of the discharge valves of the low-pressure cylinder, the resistance of the inlet valves of the high-pressure cylinder, together with the friction in the air passages and intercooler, and the reduction of air volume produced by the temperature drop in the intercooler. In well-designed compressors this lost work is small because of the use of liberal valve areas, positively operated valves, and large and properly designed intercoolers with direct and ample passages to the cylinders. If the excess work expended on the air represented by this overlapping area exceeds the saving effected by intercooling (area $JB'E'L$), no economic benefit is derived from the use of the intercooler. There may also be an unequal distribution of the work in the two cylinders and the two-stage compression may not realize a saving of power over single-stage compression for the same pressure range P_1 to P_2.

267. Performance of Air Compressors.—The *displacement* of a compressor is the volume displaced or swept through by the net area (piston area less allowance for piston rod) of the compressor piston during the compression stroke. Piston displacement is usually expressed in cubic feet per minute. For multi-stage compressors, the low-pressure cylinder only is considered.

The *capacity* of a compressor is the actual volume of air compressed and delivered, usually expressed in cubic feet of free air per minute. The capacity can be determined accurately only by actual measurement of the air output. The *volumetric efficiency* (ratio of the capacity as defined above, to the displacement of a compressor) is decreased by excessive clearance, suction friction, heating during suction and installa-

tion at high altitudes. The volumetric efficiency may be increased by reducing the temperature of the entering air, for instance, taking air from outdoors instead of from inside of a building.

Compression efficiency is the ratio of the isothermal horsepower required to compress the amount of air delivered, to the actual indicated horsepower required in the air-compressing cylinders as shown by indicator diagrams. The *mechanical efficiency* of a compressor is the ratio of the total i.hp. of the air cylinders to the steam i.hp., in the case of a steam-driven unit. For a power-driven machine the mechanical efficiency is the ratio of the air i.hp. to the brake hp. of the power drive (turbine, motor or belt). The *overall* (sometimes called *compressor*) *efficiency* is the product of the compression efficiency and the mechanical efficiency, and represents the ratio of the isothermal horsepower of the compressor to the horsepower supplied by the power drive.

268. The Air Engine.—An engine using compressed air as a medium does not differ essentially from a steam engine; in fact, a steam engine may be used without change. Air engines are often used in mines, quarries, on locomotives, tunneling, for driving small portable tools and elsewhere. For small power units, a simple form of rotary engine is sometimes employed on account of convenience irrespective of low efficiency. The absence of heat, the ease of transmission over great distances without excessive pressure loss and the absence of odor, obnoxious vapors or fumes make air a popular medium where a number of small engines are to be operated at widely scattered points. The exhausted air aids in ventilating the rooms or mine passages in which it is used. Steam, on the other hand, cannot be efficiently transmitted through long pipe lines because of the resulting condensation. Small steam engines are very inefficient and the high temperature at which they operate renders them unsuitable where handling is required.

If the expansion is complete, the cycle of the air engine is that of the compressor traversed in the opposite direction and the power delivered by the engine may be calculated by the equation for the power of the compressor, see equations 154 and 158. If the expansion is not complete, the net work done as shown by the indicator diagram can be found by integrating the areas under the individual curves. Thus if Fig. 243 represents the indicator diagram of an air engine, which has clearance

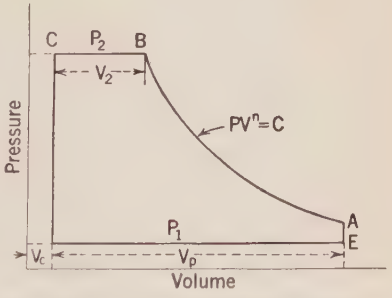


Fig. 243.—Air engine indicator diagram.

V_c and operates with incomplete expansion as shown by AE. The net work done *by* the air per cycle is

$$W = P_2(V_2 + V_c) + \frac{P_2(V_2 + V_c) - P_a(V_p + V_c)}{n - 1} - P_1V_p - P_2V_c$$

Therefore

$$W = P_2V_2 - P_1V_p + \frac{P_2(V_2 + V_c) - P_a(V_p + V_c)}{n - 1} . \quad (165)$$

where P_a is the absolute pressure at the point of release, A in Fig. 243.

Example.—Find the theoretical horsepower that can be developed by a double-acting 7-in. × 10-in. hoisting engine operating on compressed air and running at 125 r.p.m. Also estimate the air consumption in cubic feet per i.hp. per hour. The clearance is 3 per cent; temperature of inlet air, 80 deg. fahr., and pressure 75 lb. per sq. in. abs.; release pressure 39.2 lb. abs., and exhaust 14.5 lb. abs. Cut-off at 60 per cent of the stroke, and $PV^{1.32} = C$ during expansion.

Solution.—Substituting in (165),

$$W = 144\left[(75\times0.60D) - (14.5\times1.0D) + \frac{75(0.60D+0.03D) - 39.2(1.0D+0.03D)}{1.32 - 1.00}\right]$$

$$= 144[45D - 14.5D + 21.5D] = 7490D \text{ ft.-lb.}$$

Displacement is found from

$$D = \frac{\pi \times 49 \times 10}{4 \times 144 \times 12} = 0.223 \text{ cu. ft. per stroke.}$$

$$\text{(Neglecting area of piston rod) } i.hp. = \frac{7490 \times 0.223 \times 2 \times 125}{33{,}000}$$

Then $i.hp. = 12.64$.

Estimation of Air Consumption.—At point B, the total volume of air is $0.63D$. But $0.03D$ is clearance air, used repeatedly and not chargeable to the engine. However, the $0.03D$ air at 14.5 lb. abs. reduces to $0.0058D$ at 75 lb. abs. and the volume of air brought into the cylinder per stroke at 75 lb. abs. is $0.6242D$ and is per hour,

$$0.6242 \times 0.223 \times 2 \times 125 \times 60 = 2088 \text{ cu. ft.}$$

Reduced to *free air* this would be $2088 \times \frac{75 \times 520}{14.7 \times 540} = 10{,}260$ cu. ft. per hour or 812 cu. ft. of free air per i.hp. per hour.

If the air is not preheated before entering the engine cylinder, the temperature at the end of expansion is necessarily low. Thus if air enters

at 80 deg. fahr. and expands, according to $PV^{1.32} = C$, from 75 to 25 lb. per sq. in. abs. the final temperature is

$$T_a = (80 + 460)\left(\frac{25}{75}\right)^{\frac{1.32-1}{1.32}} = 414 \text{ deg. abs.}$$

or

$$t_a = -46 \text{ deg. fahr.}$$

This fall of temperature during expansion is a serious matter, for as the temperature falls the water vapor present in the air is condensed and freezes, thus causing the formation of ice in the exhaust valves and passages of the engine or air motor. This difficulty may be counteracted in various ways: by circulating warm air or water around the exhaust passages, specially designed exhaust ports, a reduced range of pressures, avoiding adiabatic and approaching isothermal expansion and by thoroughly drying the air. The only commercially successful method of avoiding inconveniently low temperatures after expansion is by raising the temperature of the inlet air. Methods of reheating are: (1) The air to be heated is passed through a coil of pipe or cast iron exposed to a fire, hot gases or steam; (2) the injection of steam or hot water into the air itself; and (3) placing an electric heating coil in the air line. The first method of heating is the most efficient and preferable from a mechanical standpoint.

Small engines, operating without expansion, have an air consumption of about 1500 to 2200 cu. ft. (115 to 170 lb.) of free air per hp. per hr. Using the air expansively in these engines reduces the air consumption to a range of 700 to 1300 cu. ft. (50 to 100 lb.), depending somewhat upon the air pressure, point of cut-off and horsepower of the engine. Figures 244 and 245 show the results of tests [1] using steam and air respectively as working mediums in an 8-in. × 18-in., 25-hp. Corliss steam engine with pressures and temperatures as noted. The curves in Fig. 245 show that the economy of an air engine is dependent upon the amount of preheating; thus with the initial temperature at 200 deg. fahr. the economy is 42 lb. of air per i.hp. per hr. at 25 hp. That quantities of mixed air and steam are capable of doing more work expansively, than equal quantities of the two media acting separately can accomplish, is evidenced by further tests [1] on the above engine. The results show that the maximum economy occurs with mixtures containing 30 per cent steam (by weight).

[1] "A Study of Air-Steam Mixtures," by L. A. Wilson and C. R. Richards. University of Illinois Experiment Station, Bulletin 131, 1922.

269. Other Uses of Compressed Air.—Aside from the driving of engines, high-pressure air is being used for a variety of purposes because of the convenience and ease of transmitting air as compared with steam.

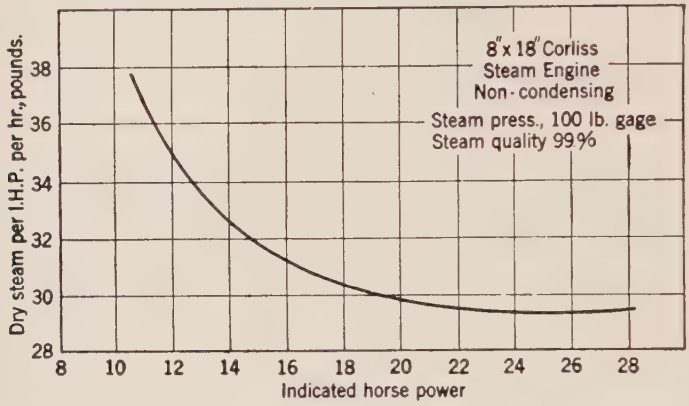


Fig. 244.—Steam consumption for 25-hp. steam engine.

Some of the important applications of compressed air are: rock drills, hammer drills, chipping, calking and scaling hammers, riveters, air hoists, lifts and jacks, sand-blast nozzles for cleaning castings and removing paint, pneumatic painting, cement guns, cleaning boiler tubes,

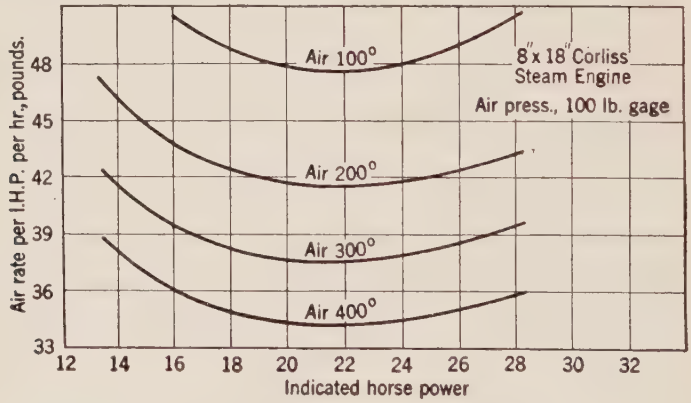


Fig. 245.—Air consumption for 25-hp. engine.

agitating liquids, pneumatic conveying systems, for pumping liquids by direct-acting pumps, by pneumatic displacement pumps, or by the ingenious air-lift system. Extensive industrial applications of compressed air have resulted in the perfecting of modern interchangeable

pneumatic tools. The gage pressure usually required for the economical operation of pneumatic tools is 70 to 90 lb. per sq. in., the pressure depending somewhat upon the type of service.

The decrease of temperature accompanying a reduction of pressure is utilized in the manufacture of liquid air. Air is compressed to about 2000 lb. pressure in a three-stage machine and discharged to a cooler. The air is next delivered to the inner tube of a long (350 ft.) double coil, through which it passes to a small orifice called an expansion valve, where the air expands into a reservoir (maintained at about 325 lb. per sq. in. gage pressure). The throttling in the expansion valve causes a drop in temperature, and the cooled and expanded air returns through the outer tube of the coil back to the compressor, cooling the descending air in the inner pipe. At each passage during this regenerative process, a decrease of temperature occurs, the effect is cumulative and ultimately the temperature in the reservoir becomes lower than the critical temperature of air (−220 deg. fahr.) and liquefaction begins.

The drop in temperature during expansion in an air engine cylinder, with the difficulties which it occasions, have been discussed in Art. 268. This low-temperature exhaust air may be utilized for refrigeration by being forced through pipes to a cold room for the cooling of its contents, the production of ice, etc. In some cases, the refrigerative-effect alone may be sought, the performance of work during expansion being incidental.

270. Measurement of Air Flow.—The quantity of air delivered by a blower or air compressor may be determined by one of the following methods: pitot tube, Venturi tube, thin-plate orifice, rounded orifice, short tube or nozzle, measuring tanks of known capacity, causing air to displace water in a tank, and recording meters. The method employing the pitot tube has been described in Chapter X. The Venturi tube has been explained in Art. 258 and illustrated by Fig. 229.

For determining the quantity of air flowing through a Venturi tube, G. A. Goodenough [1] gives

$$W = 2.056 \frac{A_2 p_1}{\sqrt{T_1}} \sqrt{\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}}$$

where W = weight of air flowing, lb. per sec.
A_2 = area of throat, sq. in.
p_1 = entrance pressure, lb. per sq. in. absolute.
p_2 = throat pressure, lb. per sq. in. absolute.
T_1 = absolute temperature of air at entrance.

[1] "Principles of Thermodynamics," Chapter IX, Third Edition, 1920, G. A. Goodenough, Holt and Co.

But $A_2 = \dfrac{\pi D_2^2}{4}$ and 1 lb. of dry air has a volume of 13.10 cu. ft. at 60 deg. fahr. and 14.7 lb. per sq. in. abs., hence the volume of free air flowing through the Venturi tube in cubic feet per minute is (when $p_2 > 0.53 p_1$)

$$Q = 2.056 \times 60 \times 13.1 \times \frac{\pi}{4} \frac{D_2^2 p_1}{\sqrt{T_1}} \sqrt{\left(\frac{p_2}{p_1}\right)^{1.43} - \left(\frac{p_2}{p_1}\right)^{1.714}}$$

Hence

$$Q = 1269 \frac{D_2^2 p_1}{\sqrt{T_1}} \sqrt{\left(\frac{p_2}{p_1}\right)^{1.43} - \left(\frac{p_2}{p_1}\right)^{1.714}} \quad . \quad . \quad . \quad . \quad (166)$$

Example.—The inlet diameter of a Venturi tube is $2\frac{1}{2}$ in. and the throat diameter $\frac{3}{4}$ in. Air flowing through the tube has an inlet temperature of 165 deg. fahr. and an inlet pressure of 54 lb. per sq. in abs.; throat pressure 30 lb. per sq. in. abs. How many cubic feet of free air are flowing per minute?

Solution.—

$$\frac{p_2}{p_1} = \frac{30}{54} = 0.556$$

$$Q = 1269 \frac{\overline{0.75}^2 \times 54}{\sqrt{625}} \sqrt{(0.556)^{1.43} - (0.556)^{1.714}}$$

Then $Q = 1544 \times 0.252 = 389.1$ cu. ft. free air per min.

The *thin-plate orifice* in the simplest and most familiar form is a round hole in a thin flat diaphragm (about 1/16 in. thick) clamped between the flanges at a joint in a pipe line, the hole being concentric with the pipe, see Fig. 246. A thicker plate is sometimes used and the plate is then chamfered around the hole on the downstream side, while the entrance side of the plate remains flat, with a sharp 90-deg. corner at the edge of the opening. Pressure connections are provided in the pipe walls on both sides of the diaphragm; so that pressure gages or manometers may be attached and the decrease of pressure through the orifice may be determined. This difference in pressure is utilized to find the rate of flow. The thin-plate orifice employs the Venturi principle, the orifice merely acting as a means whereby the fluid forms its own Venturi shape beyond the orifice plate. When the pressure connections are properly located the Venturi equation would apply, if the diameter of the minimum section (or vena contracta) of the air jet could be measured

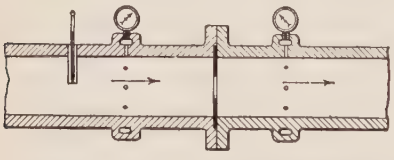


Fig. 246.—Thin-plate orifice installation.

directly. Since the diameter of the vena contracta is not directly measurable it must, in practice, be replaced by the diameter of the orifice which can be measured. For a given size orifice, the rate of flow is independent of the distance from the plane of the orifice to the plane where the static pressure is a minimum at the wall of the pipe. On the other hand, this distance varies considerably with the diameter of the orifice, and the pressure minimum is farther away for small than for large orifices. Experiments [1] have shown that the position of minimum pressure, at the vena contracta, is geometrically similar in 3-in. to 12-in. pipes for the same orifice ratios. The pressure connection on the inlet side of the orifice is usually located one pipe diameter from the orifice. The distance from the orifice to the pressure connection, at the middle of the vena contracta, on the exit side may be determined by multiplying the pipe diameter by factors given by E. G. Bailey,[1] and shown in Fig. 247.

The rate of flow through a thin-plate orifice is

$$V = C\sqrt{2gh_a}$$

in which V = velocity, feet per second.
C = orifice coefficient, see Fig. 247.
g = acceleration due to gravity, 32.17 ft. per sec.^{2}
h_a = differential pressure, feet of air.

But

$$Q = \frac{60AV}{144}, \text{ cu. ft. per minute.}$$

Hence

$$Q = \frac{60\pi D^2}{144 \times 4} C\sqrt{64.34h_a} = 2.6248CD^2\sqrt{h_a}$$

However

$$h_a = \frac{h_w \times d_w}{12 \times d_a}$$

Here h_w is the differential pressure across the orifice in inches of water, d_w and d_a are the densities in lb. per cu. ft. of water in the manometer and of the air flowing, respectively.

Therefore

$$Q = 2.6248CD^2\sqrt{\frac{d_w \times h_w}{d_a \times 12}}$$

[1] "Fluid Meters, Their Theory and Application," Part I, Second Edition, 1927, Report of A.S.M.E. Committee on Fluid Meters.

and

$$Q = 0.759CD^2\sqrt{h_w\frac{d_w}{d_a}}$$

but

$$h_w = \frac{1728p}{d_w}$$

thus

$$Q = 0.759CD^2\sqrt{\frac{1728pd_w}{d_a d_w}}$$

$$Q = 31.55CD^2\sqrt{\frac{p}{d_a}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (167)$$

in which Q = air flowing through orifice, cu. ft. per min. at exit pressure and entrance temperature.
C = orifice coefficient from Fig. 247.
D = diameter of orifice, in.
p = observed differential pressure across orifice, lb. per sq. in.
d_a = density of air, lb. per cu. ft. at exit pressure and entrance temperature, from $PV = MRT$.

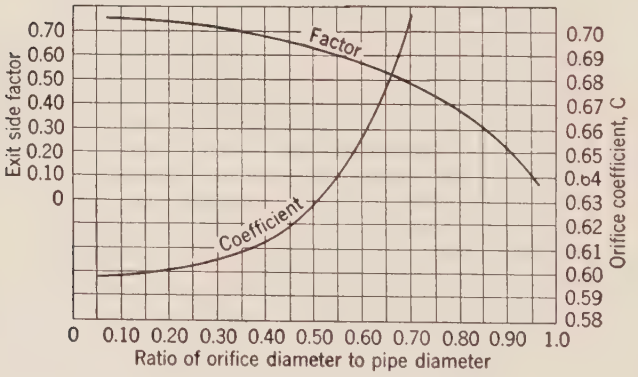


Fig. 247.—Constants for thin-plate orifices.

The orifice coefficient varies according to the ratio of the orifice diameter to pipe diameter as shown in Fig. 247. This coefficient represents the product of two factors, a frictional coefficient which allows for the frictional resistance at the entrance to the orifice, and the contraction of the stream flow which is the ratio of the area of the vena contracta to the area of the orifice.

R. J. Durley in Vol. 27 of Transactions of the A.S.M.E. under, "Air Flowing into the Atmosphere through Circular Orifices," discusses air flow through a thin-plate orifice when discharging directly to the atmosphere. The form of the equations given is similar to those stated above.

Example.—Air is flowing through a $\frac{3}{4}$-in. thin-plate orifice, the latter being inserted between the flanges of a 3-in. pipe. The pressure on the upstream side (3 in. from orifice) is 20 in. of mercury above atmosphere and the pressure on the downstream side (at the vena contracta, $2\frac{3}{16}$ in. from orifice) is 4 in. of mercury. Find the cubic feet of free air flowing per minute, if the temperature of the air on the upstream side is 140 deg. fahr. and the barometer is 29.32 in. mercury.

Solution.—

$$p = (20 - 4) \times 0.49 = 7.84 \text{ lb. per sq. in.}$$

$$C = 0.603, \text{ from curve, Fig. 247.}$$

$$d_a = \frac{M}{V} = \frac{P}{RT} = \frac{(4 + 29.32) \times 0.49 \times 144}{53.35 \times 600}$$

$$d_a = 0.0735 \text{ lb. per cu ft.}$$

Substituting in equation 167,

$$Q = 31.55 \times 0.603 \times \overline{0.75}^2 \sqrt{\frac{7.84}{0.0735}}$$

$$Q = 10.7 \times 10.34 = 110.7 \text{ cu. ft. per min. at 4 in. Hg and 140 deg. fahr.}$$

But

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

$$\frac{14.7 \times 144 \times Q_1}{520} = \frac{(4 + 29.32) \times 0.49 \times 144 \times 110.7}{600}$$

Then

$$Q_1 = 106.7 \text{ cu. ft. of free air per minute.}$$

The low-pressure nozzle test is recommended by the Compressed Air Society as an accurate method for measuring the air output of a compressor. The air delivered by the compressor is discharged into receiver A, Fig. 248, where the desired pressure is maintained constant by bleeding air through globe valve V to a second receiver B provided with a nozzle N. The diameter of B should be at least two and a half ($2\frac{1}{2}$) times the smallest nozzle diameter, preferably larger; B should

be 10 to 15 or more feet long. This nozzle is merely an orifice open to the atmosphere with curved inner edges and is of such size that the pressures in receiver B will not be more than 1 lb. per sq. in. above atmosphere, when passing all of the air through its opening. The exact pressure in B is determined by a water manometer M, and the temperature of the air flowing is measured by a thermometer T (no well) passing through the wall of receiver B, about 2 ft. from the nozzle. From these observations, together with the barometric reading, the cubic feet of air flowing through the nozzle per minute may be determined. Fig. 249 shows a sketch of the low-pressure nozzle with an accompanying table of dimensions and capacities as recommended by the Compressed Air Society. The inner surface of the nozzle must be highly polished, giving a nozzle coefficient of 0.98 to 0.99, depending upon the shape and finish.

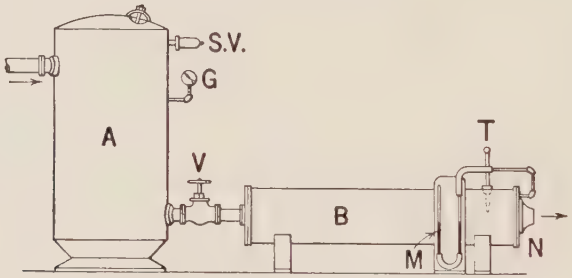


Fig. 248.—Arrangement of apparatus for air measurement by low-pressure nozzle.

When the nozzle, as selected by the table given in Fig. 249, is of such size that the manometer reading is between 3 and 20 in. of water, the quantity of air flowing through the nozzle may be calculated by equation 167, but using a new nozzle coefficient, thus

$$Q = 31.55KD^2\sqrt{\frac{p}{d_a}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (168)$$

where Q = cu. ft. of air flowing per minute at observed temperature of upstream side of nozzle and pressure of downstream side (usually atmospheric).

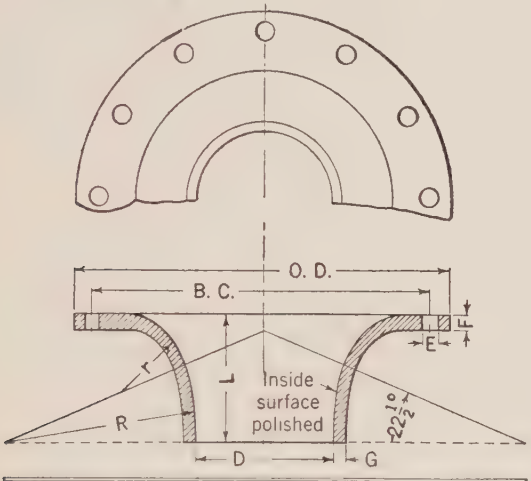
K = nozzle coefficient, 0.98 to 0.99.

D = diameter of smallest part of nozzle throat, in.

p = observed differential pressure across nozzle, lb. per sq. in.

d_a = density of air in lb. per cu. ft. at exit pressure and entrance temperature, from $PV = MRT$.

Example.—The air discharged from an air compressor flows through a 4-in. Compressed Air Society low-pressure nozzle into the atmosphere. The observed manometer reading is 12 in., the temperature of the air entering the nozzle is 206 deg. fahr., and the barometer is 29.40 in. of mercury. (*a*) How many cubic feet of free air are being compressed if the nozzle is of such shape and finish that its coefficient is 0.99? Temperature of the liquid in the



D	L	R	r	E	O. D.	B. C.	No. of bolts	Drilled for st'd flange size	Free air cu. ft. max. capacity	Free air cu. ft. min. capacity	F	G
2"	1.70"	2.8"	1 0"	⅞"	15·0"	13¼"	12	9"	340	140	⅝"	½"
3"	2.55"	4.2"	1.5"	⅞"	15·0"	13¼"	12	9"	755	290	⅝"	½"
4"	3.40"	5.6"	2.0"	⅞"	15·0"	13¼"	12	9"	1330	520	⅝"	½"
5"	4.25"	7.0"	2.5"	⅞"	15·0"	13¼"	12	9"	2065	810	⅝"	½"
7"	5.95"	9.8"	3.5"	1"	19·0"	17·0"	12	12"	3990	1560	¾"	½"
9"	7.65"	12.6"	4.5"	1⅛"	23½"	21¼"	16	16"	6600	2575	¾"	½"
11"	9.35"	15.4"	5.5"	1¼"	29½"	27¼"	20	22"	9860	3850	¾"	½"

FIG. 249.—Low-pressure nozzle details, Compressed Air Society.

manometer is 80 deg. fahr. (*b*) If the low-pressure piston displacement is 1200 cu. ft. per min., find the volumetric efficiency of the compressor.

Solution.—

$$\text{In equation 168, } p = 12 \times \frac{62.22}{1728} = 0.432 \text{ lb. per sq. in.}$$

$$d_a = \frac{M}{V} = \frac{P}{RT} = \frac{29.4 \times 0.49 \times 144}{53.35 \times 666}$$

$$d_a = 0.0584 \text{ lb. per cu. ft.}$$

Hence $$Q = 31.55 \times 0.99 \times 4^2 \sqrt{\frac{0.432}{0.0584}}$$

and $Q = 1360$ cu. ft. per min. at 206 deg. fahr. and atmospheric pressure.

But $$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$\frac{14.7 \times 144 \times Q_1}{520} = \frac{14.4 \times 144 \times 1360}{666}$$

Then $$Q_1 = 1040 \text{ cu. ft. of free air per minute.}$$

$$\text{Volumetric efficiency} = \frac{1040}{1200} \times 100 = 86.7 \text{ per cent.}$$

PROBLEMS

1. A single-stage compressor discharges 500 cu. ft. of free air per min. at 105 lb. per sq. in. abs. Inlet air temperature, 80 deg. fahr.; suction pressure 14 lb. per sq. in. abs. Find the discharge temperature, if the compression curve is $PV^{1.32} = C$.

2. In problem 1, what is the percentage of power wasted as compared with isothermal compression, the cycle being like *ABCD*, Fig. 239?

3. In problem 1, what is the saving of power effected by the water jacket?

4. Assuming the clearance in problem 1 to be 4 per cent, find the cylinder volume in cubic feet, if the compressor is double-acting and makes 100 r.p.m.

5. Find the power required to compress 1000 cu. ft. of free air per minute from 14.5 lb. abs. to 110 lb. abs. in a single-stage compressor; $n = 1.35$, clearance $2\frac{1}{2}$ per cent and inlet air temperature 80 deg. fahr.

6. Find the power required to compress 1000 cu. ft. of free air per minute from 14.5 lb. abs. to 110 lb. abs. in a two-stage compressor with perfect intercooling. $n = 1.35$, clearance $2\frac{1}{2}$ per cent and inlet air temperature 80 deg. fahr. Find the intermediate pressure.

7. Find the cylinder dimensions of the compressors in problems 5 and 6 if they operate at 120 r.p.m. and are double-acting. Piston speed to be 500 ft. per min. in each case. Assume 2-lb. pressure drop in the intercooler of the two-stage compressor.

8. The clearance of an air compressor is 3.2 per cent and the initial and final pressures are respectively 14.4 lb. and 72 lb. abs. Find the volumetric efficiency if $n = 1.30$. What would be the volumetric efficiency if the final pressure were 155 lb. absolute?

9. An air compressor with a cylinder 12 in. $\times$ 18 in. is double-acting and driven at a speed of 110 r.p.m. Clearance is 3 per cent, $n = 1.33$; $p_1 = 14.5$ lb. abs. and $p_2 = 89.5$ lb. abs. Find (*a*) the capacity of the compressor in cubic feet per minute; (*b*) the indicated horsepower of the steam engine driving the compressor if the mechanical efficiency is 88 per cent.

10. Find the heat removed in the intercooler for the compressor of problem 6. Find the theoretical amount of water required per minute, if the water available is at 58 deg. fahr. and is to leave at 90 deg. fahr. In the actual intercooler would more or less water be required? Why?

11. For a given amount of air entering the compressor, find the relative work for single-stage, two-stage and three-stage compression between 14 lb. and 280 lb. abs., take $n = 1.3$.

12. Find the size of a double-acting air engine to develop 40 hp. when running at 300 r.p.m. Clearance 2 per cent, temperature of inlet air 80 deg. fahr., and pressure 150 lb. abs. Cut-off at 40 per cent of the stroke, and exhaust 14.2 lb. abs. Piston speed 400 ft. per min. and $PV^{1.3} = C$.

13. What is the temperature of the air at the end of expansion in the engine of problem 12, that is, at 20 lb. abs. pressure? If the air is supplied at 420 deg. fahr., what is the temperature after expansion to 20 lb. abs. pressure?

14. Air flowing through a Venturi tube has an inlet temperature of 112 deg. fahr. and an inlet pressure of 10 in. of mercury gage, throat pressure 6 in. of mercury gage and barometer reading is 29.24 in. of mercury. Inlet diameter of Venturi is 4 in. and the throat diameter $1\frac{1}{4}$ in. How many cubic feet of free air are flowing per minute?

15. A 3-in. thin-plate orifice is placed between the flanges of an 8-in. pipe line, through which air is flowing at 150-lb. gage pressure. The pressure on the downstream side at the vena contracta is 50-lb. gage, barometer 29.72-in. mercury. Find the cubic feet of free air flowing per minute, if the temperature of the air on the upstream side is 90 deg. fahr. Where will the downstream pressure connection be located?

16. The discharge from an air compressor at a constant pressure of 100-lb. gage flows through a 7-in. Compressed Air Society nozzle into the atmosphere. The observed manometer reading is 10 in. of water, temperature of liquid in the manometer 60 deg. fahr., temperature of air entering nozzle 132 deg. fahr. and the barometer is 29.38 in. of mercury. How many cu. ft. of free air are being discharged, if the nozzle coefficient is 0.99?

CHAPTER XIX

INTERNAL-COMBUSTION ENGINES

271. Description and Operation.—The heat generated by the combustion of fuel in a furnace passes through the metal plates or tubes of the boiler to the water, and the steam generated may be used in an engine cylinder or turbine to do work. The working medium is therefore entirely different from the products of combustion of the fuel. For the production of power with steam there are generally four different parts: the furnace, boiler, engine or turbine, and condenser.

In an internal-combustion engine the fuel, which is to supply heat-energy for conversion into work, forms part of the working substance. Combustion takes place within the cylinder and work is done by expansion of the resulting gases. The working substance therefore, undergoes a chemical change during this operation and the thermodynamic process is not cyclic. In the early stages, before combustion, the substance is a mixture of fuel with air, generally an excess of air is required to provide enough oxygen for complete combustion. In the later stages, after combustion, there is a mixture of the products of combustion with nitrogen and any surplus of air. The fuel commonly enters as a gas or vapor and is drawn in with a suitable proportion of air; but the fuel may be injected into the cylinder as a liquid, becoming vaporized after admission or being burned as it enters. As a rule the only chemically active constituents of the fuel are hydrogen, hydrogen-carbon compounds, and carbon monoxide. During combustion these constituents unite with oxygen to form water-vapor and carbon dioxide. The nitrogen of the air takes no part in the chemical process beyond acting as a diluent.

Typical examples of internal-combustion engines are the ordinary gas and gasoline engines, in which a " charge " of air mixed with combustible gas or vaporized liquid fuel is drawn into the cylinder by the piston. The mixture is compressed into a clearance space, and there ignited by an electric spark, so that explosive combustion takes place while the volume of the charge is nearly constant. The heat thus internally developed gives the working substance a high temperature and pressure; expansion of the gases occurs, and work is done as the piston advances.

Considered thermodynamically, internal-combustion engines have the advantage, over the steam engine and turbine, that the working substance "takes in" heat (by its own combustion) at a very high temperature. In the combustion of the charge a temperature of 4000 deg. fahr. or more is reached. The full thermodynamic advantages of a high initial temperature cannot be realized in practice, for if the cylinder walls were allowed to reach this high temperature they would soon be destroyed. Lubrication of the piston would also be impossible, to say nothing of the probable premature ignition of the incoming fuel mixture; hence the cylinder is generally water-jacketed to keep the walls and parts cool. With large gas engines the difficulty is to keep the cylinder, piston and valves cool; with steam engines, the cylinder should be kept hot to reduce the losses due to condensation of the steam. The average temperature at which the heat is developed in an internal-combustion engine is far above that at which heat is received by the working medium of a steam engine or turbine. On the other hand, internal-combustion engines do not discharge heat at as low a temperature as do steam engines and turbines. But the actual working range of temperature is so large that a gas engine can convert into work a larger fraction of the heat-energy of the fuel than is utilized by any steam engine or steam turbine. A good internal-combustion engine will convert about 35 to 40 per cent of the energy of the fuel into work; the best steam engines convert no more than about 20 per cent, and the best steam turbines convert no more than about 30 per cent of the energy of the fuel into work.

The small internal-combustion engine has made possible the automobile, tractor and airplane. In plants having combustible by-product gases, such as those from blast furnaces and coke ovens, large-size units are frequently used. Internal-combustion engines are well adapted for use in marine service and small plants, where space is of prime importance.

272. The Four-Stroke Cycle.—In the most usual type of internal-combustion engine the mechanical cycle is completed in four strokes of the piston or two revolutions of the crankshaft. (1) The charge of gas and air is drawn into the cylinder D, Fig. 250, inlet valve E being open on the outward stroke of the piston C. (2) On the return stroke of C, inlet valve E and exhaust valve F being closed, the mixture of air and gas is compressed into the clearance space. Near the head-end dead center, the compressed mixture is ignited by an electric spark, and burns; the pressure suddenly increases. (3) During the next forward stroke the fired mixture expands, doing work on piston C which through B creates a turning effort of crankshaft A. (4) As piston C moves to the left on the

fourth stroke, the products of combustion are forced out of the cylinder, exhaust valve F being open. A small quantity of the burnt mixture remains in the clearance space, and is mixed with the next charge, unless special means are employed to remove these inert and non-combustible gases by "scavenging."

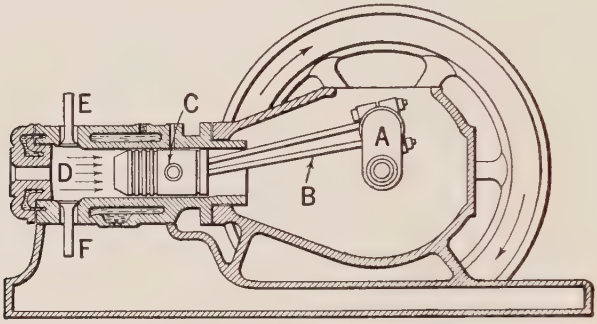


Fig. 250.—Diagram of a four-stroke Otto cycle engine.

The indicator diagram of the four-stroke cycle is shown diagrammatically by Fig. 251. During the suction stroke 0–1 the pressure is below atmospheric and during exhaust 4–0 the pressure is above atmospheric. The area of this loop, 0–1–5, represents the work done *on* the gases during the cycle and is due to the "pumping action" of the piston in forcing out the exhaust gases (high pressure at 4) and drawing in the charge during suction (low pressure along 0–1). The curve 1–5–2 represents compression of the charge, V_c being the volume of the clearance space into which the charge is compressed. Line 2–3 is the rise of pressure caused by the explosion, and 3–4 is the expansion which constitutes the effective working stroke. The area 2–3–4–5 represents work done *by* the gases during the cycle, and the net work developed during the four-stroke cycle is therefore (2–3–4–5) minus (0–1–5).

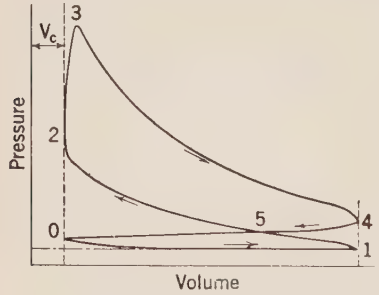


Fig. 251.—Indicator diagram of the Otto four-stroke cycle.

273. The Two-Stroke Cycle.—Fig. 252 illustrates the action of an internal-combustion engine in which the completion of one cycle of operation is accomplished in two strokes of the piston. At A the scavenging valves in the cylinder head are open, and a charge of pure

air (compressed by an air compressor) blows into the cylinder, forcing out the exhaust gases as shown. The scavenging valves close, and as the piston moves upward the exhaust ports are covered, Fig. 252*B*.

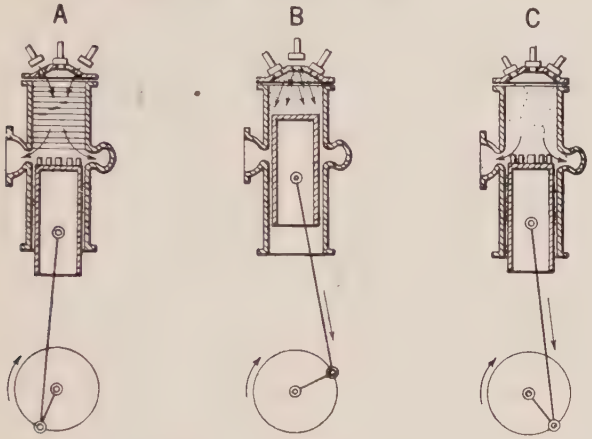


FIG. 252.—Diagram of the Diesel two-stroke cycle engine.

Continued upward motion compresses the air charge until upper dead center is reached. The clearance volume is very small and the maximum or final compression pressure rises to about 500 lb. per sq. in. The work done on the air charge during compression causes the temperature to rise to 1000 deg. fahr. or more. Just before the piston reaches top dead center, Fig. 252*B*, a charge of fuel is injected into the cylinder and combustion occurs as the piston starts downward. The piston continues to the end of its stroke under the influence of the expanding gases. Before the completion of the downward stroke, Fig. 252*C*, the piston uncovers passages or ports in the side of the cylinder through which the exhaust gases pass. By this means one explosion is obtained every revolution of the engine shaft when using one single-acting cylinder.

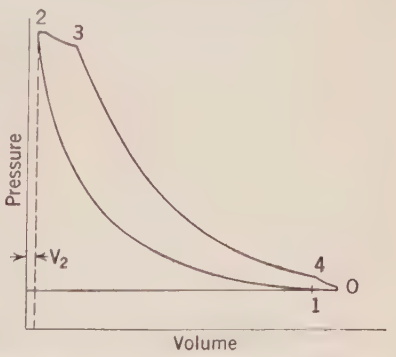


FIG. 253.—Indicator diagram of the Diesel two-stroke cycle.

An indicator diagram of a two-stroke cycle engine is shown by Fig. 253. A charge of pure air is admitted under pressure from 0–1; compression of this air with a resultant high pressure and temperature occurs along 1–2. Injection of fuel and combustion at nearly constant pres-

sure is represented by line 2–3. Expansion of the gases takes place along 3–4. From 4 to 0, exhaust occurs and scavenging of the cylinder is effected.

274. Cycles of Combustion.—The principal combustion cycles used in modern internal-combustion engines are the *Otto* and *Diesel* cycles. In the Otto cycle, Fig. 251, the events are: the mixture of gas or fuel-vapor and air drawn into the cylinder, compression of gas and air, then combustion at nearly constant volume, next expansion and finally exhaust. An engine operating on the Diesel cycle, Fig. 253, functions as follows: during the suction stroke air alone is drawn into the cylinder, compression of the air, then injection of the fuel with combustion at nearly constant pressure, next expansion, and finally exhaust. The Otto cycle may be employed in engines using the two-stroke or four-stroke principle with gaseous fuels or for liquid fuels (such as gasoline) that can be readily vaporized before entering the cylinder. The Diesel cycle employs the two-stroke or four-stroke principle with liquid fuels which may be injected as a liquid (solid-injection) or as a fuel and air mixture (air-injection).

An engine may be described as: Otto cycle, 60 b.hp., 2000 r.p.m., six-cylinder, single-acting, vertical, four-stroke, gasoline (carburetor), or automobile. Another engine can be properly identified thus: Diesel, 240 b.hp., 375 r.p.m., four-cylinder, double-acting, vertical, two-stroke, fuel oil (solid-injection), or marine.

275. The Otto Cycle.—The indicator diagram of the ideal Otto cycle is shown by Fig. 254. The line 0–1 (atmospheric pressure) indicates the entrance of the charge of fuel and air during the first stroke, the curve 1–2 adiabatic compression of this charge on the return stroke. The line 2–3 represents the rise of pressure at constant volume when the charge is ignited and the fuel burned, and the curve 3–4, adiabatic expansion of the products of combustion. At point 4 (end of the expansion stroke) the exhaust valve opens with the result that part of the gas at once escapes and the pressure falls to that of the atmosphere, giving line 4–1. Line 1–0 represents the exhaust stroke by which the cylinder is emptied of burned gases, preparatory to receiving a fresh charge during the first stroke of the next cycle. If the ideal engine were of the two-stroke cycle type, line 0–1 would be omitted from the indicator diagram.

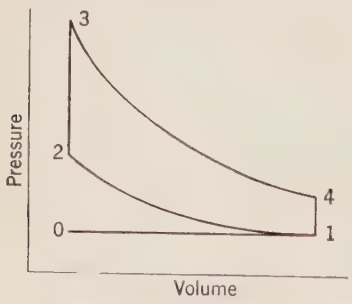


Fig. 254.– Indicator diagram of the ideal Otto cycle.

For the purpose of obtaining a simple standard with which real engines may be compared, a practice has been adopted of treating the working substance as air, to which between 2 and 3, Fig. 254, there is imparted a definite quantity of heat, namely, the heat of combustion of the charge. This standard is known as the " Air Standard " and is a measure of the ideal thermal efficiency of an internal-combustion engine. Such an ideal efficiency affords a useful standard for comparison with the performance of a real engine. To simplify the analysis, certain conditions are assumed for this ideal air standard Otto cycle:

(1) That the mixture of gas and air is admitted during the suction stroke at atmospheric pressure.
(2) That compression is adiabatic, which necessitates the assumption that the cylinder walls and piston are non-conductors of heat.
(3) That combustion is instantaneous and complete, and takes place at constant volume.
(4) That expansion is adiabatic.
(5) That the products of combustion are exhausted at atmospheric pressure, and no exhaust gases left in the clearance space.
(6) That the specific heat is constant throughout the cycle (independent of temperature as well as pressure).
(7) That in the case of a liquid fuel, the fuel is completely vaporized before entering the cylinder.

The lines 1–2 and 3–4, Fig. 254, are assumed to be adiabatics. All the heat must therefore be supplied along line 2–3 and rejected along 4–1. The *heat supplied* along 2–3 is

$$ {}_2Q_3 = Mc_v(T_3 - T_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (169) $$

in which M = weight of the charge, lb.
c_v = specific heat at constant volume, 0.1739 for air.
T_3 = absolute temperature at 3, deg. fahr.
T_2 = absolute temperature at 2, deg. fahr.

The *heat rejected* along 4–1 is

$$ {}_4Q_1 = Mc_v(T_4 - T_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (170) $$

in which T_4 = absolute temperature at 4, deg. fahr.
T_1 = absolute temperature at 1, deg. fahr.

The net work done is

$$ {}_2Q_3 - {}_4Q_1 = Mc_v(T_3 - T_2) - Mc_v(T_4 - T_1) $$

The efficiency is:

$$\frac{{}_2Q_3 - {}_4Q_1}{{}_2Q_3} = \frac{Mc_v(T_3 - T_2) - Mc_v(T_4 - T_1)}{Mc_v(T_3 - T_2)}$$

$$= \frac{(T_3 - T_2) - (T_4 - T_1)}{T_3 - T_2}$$

$$e_a = 1 - \frac{T_4 - T_1}{T_3 - T_2} \quad \ldots\ldots\ldots \quad (171)$$

Then by Art. 16,

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} \quad \text{and} \quad \frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{n-1}$$

But $V_3 = V_2$ and $V_4 = V_1$ as shown in Fig. 254.
Hence

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} \quad \text{and} \quad \frac{T_4}{T_3} = \left(\frac{V_2}{V_1}\right)^{n-1}$$

and

$$\frac{T_1}{T_2} = \frac{T_4}{T_3} \quad \text{or} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Then

$$\frac{T_4}{T_1} - 1 = \frac{T_3}{T_2} - 1 \quad \text{and} \quad \frac{T_4 - T_1}{T_1} = \frac{T_3 - T_2}{T_2}$$

or

$$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_1}{T_2} \quad \ldots\ldots\ldots\ldots\ldots \quad (172)$$

Substituting equation 172 in equation 171,

$$e_a = 1 - \frac{T_1}{T_2} \quad \ldots\ldots\ldots\ldots\ldots \quad (173)$$

But

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \quad \text{also} \quad \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} \text{ (see Art. 16)}$$

Therefore

$$e_a = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \quad \ldots\ldots\ldots\ldots \quad (174)$$

And

$$e_a = 1 - \left(\frac{V_2}{V_1}\right)^{n-1} \quad \ldots\ldots\ldots\ldots \quad (175)$$

Equation 173 shows that the air standard thermal efficiency of a gas engine employing the Otto cycle depends upon the ratio of the absolute temperatures before and after compression. From equation 175, the smaller the clearance, V_2, of the engine, the larger is the efficiency. Equation 174 is important as showing the beneficial influence of compression, the efficiency increasing as P_2 increases, since the pressure P_1 remains nearly constant.

Example. —In an engine operating on the ideal Otto cycle the suction pressure is 14.7 lb. per sq. in. abs., temperature at the beginning of compression 140 deg. fahr., highest pressure attainable (at the end of combustion) 350 lb. per sq. in. abs. at a temperature of 2140 deg. fahr. Find: (*a*) the pressure at the end of compression, (*b*) the temperature at the end of compression, (*c*) the air standard thermal efficiency, and (*d*) the clearance volume in per cent of the stroke. Assume $n = k = 1.40$.

Solution.—

(*a*) From equation 36, $\dfrac{T_2}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}}$

$$T_2 = 600\left(\frac{P_2}{14.7}\right)^{\frac{0.4}{1.4}} \qquad \text{(A)}$$

From equation 16, $\dfrac{P_2V_2}{T_2} = \dfrac{P_3V_3}{T_3}$ and $V_2 = V_3$, see Fig. 254.

Then

$$T_2 = 2600\frac{P_2}{350} \qquad \text{(B)}$$

Putting T_2 of (A) in place of T_2 in (B),

$$600\left(\frac{P_2}{14.7}\right)^{\frac{2}{7}} = 2600\left(\frac{P_2}{350}\right)$$

Hence $P_2 = (37.44)^{1.4} = 160$ lb. per sq. in. abs.

(*b*) $\quad T_2 = 2600\left(\dfrac{160}{350}\right) = 1188$ deg. abs. or 728 deg. fahr.

(*c*) $\quad e_a = 1 - \dfrac{T_1}{T_2} = 1 - \dfrac{600}{1188} = 1 - 0.505 = 0.495$ or 49.5 per cent.

(*d*) From equation 35, $P_1V_1^n = P_2V_2^n$, therefore $\dfrac{P_2}{P_1} = \left(\dfrac{V_1}{V_2}\right)^n$

Take the piston displacement = 1.00, then $V_1 = V_2 + 1$.

Thus $\qquad \dfrac{160}{14.7} = \left(\dfrac{V_2 + 1}{V_2}\right)^n$

and $V_2 = 0.222$ or 22.2 per cent of the stroke.

With increased ratios of pressures $\left(\frac{P_2}{P_1}\right)$ the air standard efficiency of the ideal Otto cycle increases as shown in Table XIII, taking $n = k = 1.4$.

TABLE XIII

Effect of Compression on Efficiency of Ideal Otto Cycle

P_1 in Lb. per Sq. In. Abs.	P_2 in Lb. per Sq. In. Abs.	$\frac{P_2}{P_1}$	Air Standard Thermal Eff.	Clearance Required
14.7	29.4	2	0.179	
	58.8	4	0.327	0.591
	88.2	6	0.401	0.386
	117.6	8	0.448	0.293
	147.0	10	0.483	0.239
	176.4	12	0.508	0.204
	205.8	14	0.530	0.179
	235.2	16	0.547	0.160
	294.0	20	0.575	0.133
	441.0	30	0.622	0.097

Table XIII shows that there is at first a very rapid gain of efficiency with increased compression, but when the compression is high the thermodynamic advantage of increasing the pressure is small. The efficiencies actually obtained in tests of engines are considerably lower, usually 40 to 60 per cent less, owing to heat losses, variation in specific heat, and to the fact that combustion is not instantaneous.

In all engines there is a practical limit to the amount of compression. The pressure must not be so great as to cause pre-ignition by unduly raising the temperature of the charge before the end of the compression stroke. The compression pressures for engines operating on the Otto cycle vary from 40 to 200 lb. per sq. in. differing with various kinds of fuels; for example, being low when there is much hydrogen and comparatively high for alcohol. For gasoline automobile engines, the pressures range from 60 to 100 lb. per sq. in. Difficulty may be experienced with automobile engines designed for the higher compression pressures when operated with ordinary gasoline. With producer-gas as a fuel, compression pressures of 100 to 150 lb. per sq. in. are allowable; and with blast-furnace gas, pressures as high as 180 lb. per sq. in. may be used.

Comparison of Fig. 251 and Fig. 254 shows the variation of the actual Otto cycle from the theoretical. In the actual engine, combustion is not

instantaneous but begins before the piston reaches the end of the compression stroke, at 2, in Fig. 251. The exhaust valve is generally opened before the end of the expansion stroke, in order that the pressure in the cylinder may be practically atmospheric before the piston starts on the exhaust stroke. The exhaust stroke, Fig. 251, is at a pressure slightly above atmospheric and during the suction stroke the pressure in the cylinder is below atmospheric. The area of this loop, representing a loss of work, is usually small, especially if ample valve areas are provided and the engine is operated at a relatively low speed. The compression and expansion curves of the actual card are not true adiabatics, principally because of the heat loss to the water-jacket and variable specific heat; the curves, however, approximate very closely the polytropic curve $PV^n = C$. For a definite value of n, the discussion and equations given in this article apply for any gas. The value of n for gasoline mixtures is generally taken as 1.32 for both the compression and expansion curves.

276. The Diesel Cycle.—The ordinary gas engines, automobile engines and many oil engines are of the constant-volume (Otto) type of internal-combustion engine. If the pressure of the working substance does not change while combustion is taking place, the cycle is called the constant-pressure or Diesel cycle. Suppose that the air is separately compressed into the clearance space before any fuel is admitted and that the fuel (oil) is then forced into the cylinder, burning as it enters, while the piston begins its outward (expansion) stroke. By suitably regulating the rate of admission of the fuel, the pressure may be kept constant until combustion is completed, 2–3, Fig. 255. By avoiding any admixture of fuel with the air in the cylinder, before and during compression, a high ratio of compression is practicable with the Diesel cycle with no possibility of pre-ignition, and consequently a high thermal efficiency is generally obtained.

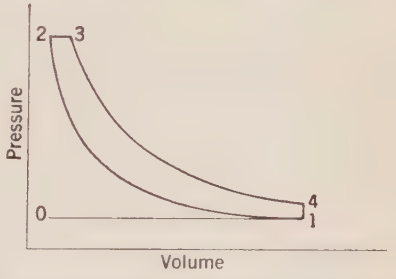


Fig. 255.—Indicator diagram of the ideal Diesel cycle.

In the ideal Diesel engine cycle, Fig. 255, air enters the cylinder along 0–1, the air is compressed on the return stroke of the piston along the adiabatic curve 1–2. The air is compressed to about 500 lb. per sq. in. or more before the fuel is admitted. At 2 the fuel is forced into the cylinder and, due to the high temperature of the compressed air, the fuel ignites. The supply of fuel continues as the piston moves back, causing combustion at constant pressure. The curve 3–4 represents the adia-

batic expansion of the products of combustion. At point 4 the exhaust valve opens and the pressure drops to 1 (atmospheric). On the fourth stroke, along 1–0, the products of combustion are forced through the exhaust opening; some of the products, however, remain in the clearance space and are mixed with the incoming air on the first stroke of the next cycle.

To simplify the analysis, certain conditions are assumed for the ideal air standard cycle of the Diesel engine:

(1) That the pressure of the air during the suction stroke is atmospheric.
(2) That compression is adiabatic, which necessitates the assumption that the cylinder walls and piston are non-conductors of heat.
(3) That the combustion of the fuel is at constant pressure, that is, the line 2–3, Fig. 255, is horizontal.
(4) That in the case of liquid fuel, the fuel is completely vaporized before entering the cylinder.
(5) That the fuel is injected into the cylinder without admixture of air, and that it enters the cylinder at the temperature T_1.
(6) That expansion is adiabatic.
(7) That the products of combustion are exhausted at atmospheric pressure, and no exhaust gases are left in the clearance space.
(8) That the specific heat is constant throughout the cycle (independent of temperature as well as pressure).

Since the lines 1–2 and 3–4, Fig. 255, are assumed adiabatics, all the heat of the cycle must therefore be supplied along line 2–3 and rejected along 4–1. The *heat supplied along* 2–3 is

$$ {}_2Q_3 = Mc_p(T_3 - T_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad (176) $$

in which M = weight of the air in the cylinder, assumed as constant since the fuel is oil and the increase in weight of the charge is small.
c_p = specific heat at constant pressure, 0.2425 for air.
T_3 = absolute temperature at 3, deg. fahr.
T_2 = absolute temperature at 2, deg. fahr.

The *heat rejected* along 4–1 is

$$ {}_4Q_1 = Mc_v(T_4 - T_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad (177) $$

in which c_v = specific heat at constant volume, 0.1739 for air.
T_4 = absolute temperature at 4, deg. fahr.
T_1 = absolute temperature at 1, deg. fahr.

The net work done is

$$_2Q_3 - {}_4Q_1 = Mc_p(T_3 - T_2) - Mc_v(T_4 - T_1)$$

The efficiency is

$$\frac{{}_2Q_3 - {}_4Q_1}{{}_2Q_3} = \frac{Mc_p(T_3 - T_2) - Mc_v(T_4 - T_1)}{Mc_p(T_3 - T_2)}$$

$$e'_a = 1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (178)$$

Since 2–3 is at constant pressure and 4–1 at constant volume

$$T_3 = T_2\left(\frac{V_3}{V_2}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (179)$$

and

$$T_4 = T_1\left(\frac{P_4}{P_1}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (180)$$

As 1–2 and 3–4 are adiabatics

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n \quad \text{and} \quad \frac{P_4}{P_3} = \left(\frac{V_3}{V_4}\right)^n$$

By dividing and since $V_1 = V_4$

$$\frac{P_4}{P_1} = \left(\frac{V_3}{V_2}\right)^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (181)$$

Substituting (181) in (180)

$$T_4 = T_1\left(\frac{V_3}{V_2}\right)^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (182)$$

Substituting (182) and (179) in (178),

$$e'_a = 1 - \frac{T_1}{T_2}\left\{\frac{\left[\left(\frac{V_3}{V_2}\right)^n - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)}\right\} \quad . \quad . \quad . \quad . \quad (183)$$

Since T_1 and T_2 are unknown in most cases a ratio of volumes and pressures will usually be more convenient for finding the air standard efficiency of the Diesel engine cycle. Thus

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$$

Hence

$$e'_a = 1 - \left(\frac{V_2}{V_1}\right)^{n-1} \left\{ \frac{\left[\left(\frac{V_3}{V_2}\right)^n - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)} \right\} \quad . \quad . \quad . \quad . \quad (184)$$

And

$$e'_a = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \left\{ \frac{\left[\left(\frac{V_3}{V_2}\right)^n - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)} \right\} \quad . \quad . \quad . \quad . \quad (185)$$

Equation 185 shows that the thermal efficiency of the Diesel cycle depends not only upon the compression, but also upon the ratio of the volume (V_3) at cut-off to the clearance volume (V_2). The efficiency increases as this ratio decreases.

Example.—In an engine operating on the ideal Diesel cycle the suction pressure is 14.5 lb. per sq. in abs., and the fuel is injected for 4 per cent of the stroke. The clearance volume is 6 per cent of the stroke. Find the air standard thermal efficiency, assuming $n = k = 1.4$.

Solution.—From equation 184,

$$e'_a = 1 - \left(\frac{0.06}{1.06}\right)^{0.4} \left\{ \frac{\left[\left(\frac{0.10}{0.06}\right)^{1.4} - 1\right]}{1.4\left(\frac{0.10}{0.06} - 1\right)} \right\}$$

$$e'_a = 1 - \frac{1.05}{3.15 \times 0.933}$$

$$= 0.642 \text{ or } 64.2 \text{ per cent}$$

Comparison of Figs. 253 and 255 shows the deviation of the actual card from the theoretical. In the actual engine, compression approaches the adiabatic curve since practically pure air is compressed in the cylinder, and the value of exponent n is usually between 1.35 and 1.40. In general, the fuel cannot be injected rapidly enough to maintain a constant pressure and line 2–3, Fig. 253, shows a slight decrease during combustion of the fuel. The value of n for the expansion curve is about 1.35 or less.

277. Losses in Internal-Combustion Engines.—The total amount of heat supplied during an internal-combustion engine cycle may be accounted for as follows:

1. Heat converted into work.
2. Heat used in overcoming friction.
3. Heat carried away in the exhaust gases.
4. Heat given to the water-jacket.
5. Heat lost by radiation, leakage, etc.

There are wide variations in the relative proportions of the above losses depending upon the type, size and operating conditions of the engine under consideration. For an automobile engine operating on the Otto cycle, the distribution of heat may be: heat converted into work about 20 per cent, lost to friction 5 per cent, carried away in the exhaust gases 43 per cent, given to the water-jacket 22 per cent, and radiation and other losses 10 per cent. For Diesel engines, the distribution of heat may be: converted into work about 30 per cent, lost to friction 12 per cent, carried away in exhaust gases 28 per cent, given to water-jacket 25 per cent, and radiation and other losses 5 per cent.

The heat carried away in the exhaust gases is generally the largest loss, resulting from incomplete expansion and exhaust at a high temperature (400 to 1200 deg. fahr.). This unavoidable loss can be reduced by carrying the expansion down to a lower pressure, but mechanical difficulties are encountered. The high temperature attained during explosion results in a very rapid extraction of heat by the cylinder walls. The higher the temperature the more rapid is this loss through the cylinder walls to the water-jacket. Cooling is necessary as a protection for the cylinder walls, to facilitate lubrication, and to prevent preignition. Because of the higher pressures employed in Diesel engines the moving parts are heavier than in Otto cycle engines of equal capacity, and hence the frictional losses are higher in Diesel engines.

278. Fuels.—Internal-combustion engines may operate on any combustible fuel, gaseous or liquid. The more expensive fuels are seldom used. Ordinary city gas is commonly a mixture of coal or oil gas and water gas, and as commonly produced is too expensive for extensive use in gas engines. City gas varies widely in its composition, depending upon how it is made, and usually contains about 40 per cent H_2, 35 per cent marsh gas, CH_4, and varying amounts of CO and olefiant gas, C_2H_4. The heating value averages 500 to 600 B.t.u. per cubic foot.

Where natural gas is plentiful and cheap, it is used as a gas-engine fuel. Natural gas is composed mainly of marsh gas, CH_4. In some natural gases, there is considerable free hydrogen and also an appreciable amount of olefiant gas, C_2H_4. The heating value is usually between 850 and 1100 B.t.u. per cubic foot.

At some blast-furnace plants, the gas from the furnace is used for

fuel in the gas engines that drive the blowers. The principal combustible in blast-furnace gas is CO, which averages 30 per cent. The heating value of blast-furnace gas is about 100 B.t.u. per cubic foot.

The by-product gas from coke ovens is being used extensively as a gas-engine fuel. Coke-oven gas contains about 50 per cent H_2 and 35 per cent CH_4, and has a heating value of 400 to 500 B.t.u. per cubic foot.

In a gas producer, coal is burned with a deficiency of air and the gas formed contains about 15 per cent H_2 and 25 per cent CO. Producer gas has a heating value of 100 to 150 B.t.u. per cubic foot.

The liquid fuels used in internal-combustion engines vary from crude oil to more refined oil products, such as gasoline. The heating value of these liquid fuels is generally from 18,000 to 20,000 B.t.u. per pound. Crude oil is not used to any great extent as it contains valuable light fuels which can be distilled off, leaving behind a fuel-oil residue. The fuel oil is generally used by employing air-injection, solid-injection, or a vaporizer which forms a part of the clearance space, explained in Art. 281. Gasoline is the name given to a fuel derived from crude oil (petroleum) and suitable for use in an automobile engine. Vaporization of gasoline is readily accomplished by spraying it into a current of air, as is done in the gasoline carburetor.

279. The Gasoline Carburetor.—There are a great variety of carburetors or mixing valves. Only one simple carburetor will be described here, but the fundamental principle is the same for all. The carburetor is a device that is intended to give the correct mixture of air and fuel at all speeds and loads on the engine. Better results are obtained with complete vaporization of the fuel, but evaporation of the fuel is not necessary if the liquid particles are finely divided and thoroughly mixed with the air.

Fig. 256 shows the parts of a common type of carburetor. The float operates the needle valve, thus maintaining the gasoline at a constant level in the float chamber. When the engine is not in operation the level of the gasoline in the idling tube and accelerating well will be the same as that in the float chamber. The fuel in the accelerating well is about $\frac{1}{16}$ in. below the gasoline spray nozzles, to prevent loss of fuel when the engine is idle. When the engine is running, at each suction stroke of the piston, air is drawn in through the air inlet and passes at a high velocity through the small Venturi tube and over the gasoline spray nozzles. The flow of gasoline from the float chamber is regulated by the high-speed adjustment needle, from which the gasoline flows into the bottom of the idling tube and accelerating well, both of which are in communication with the air-bleeder valve. The gasoline level at the air bleeder is

the same as in the float chamber. The air bleeder admits a small amount of air into the gasoline passages for the purpose of breaking up the gasoline into a mist as it discharges from the small Venturi (expanding tube) into the large Venturi (mixing tube). By the use of this small amount of air, complete vaporization in the mixing chamber can be made more easily.

The accelerating well and the passages just above it are intended to furnish the extra amount of gasoline needed when the throttle valve is

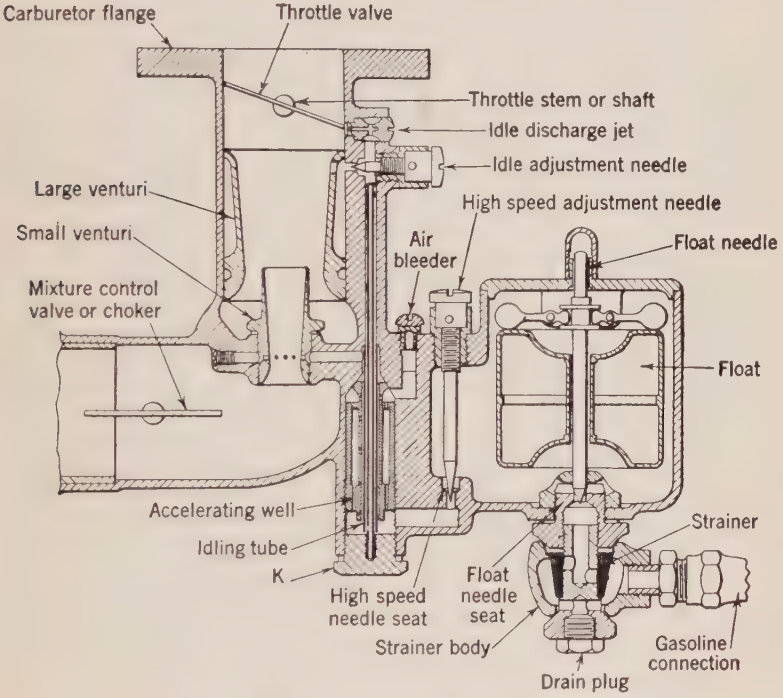


Fig. 256.—Section of a Stromberg model "M" carburetor.

suddenly opened. The idling tube, controlled by the idle adjustment needle, is in the center of the main gasoline supply passage (accelerating well) and is intended to supply a small amount of gasoline to the engine through the idle discharge jet when the throttle valve is nearly closed. In the operation of this idling tube some air is drawn through a restricted opening under the control of the idle adjustment needle. This air mixes with the gasoline and the mixture is discharged as a spray through the idle jet. When the throttle valve is more than half open there is insufficient suction to discharge a gasoline spray through the idle jet, and all

of the spray enters from the holes opening into the small Venturi or expanding tube.

Experiments have shown that as the velocity of the air through the Venturi tube becomes greater (as the throttle valve opens) the flow of gasoline increases at a faster rate than does the air. If the nozzles were adjusted to give correct proportions at any given velocity of air through the Venturi tube, the mixture would be too rich at higher air velocities and too lean at lower velocities. Some of the methods employed to compensate for the tendency of the mixture to change with variation in engine speed are: (1) additional fuel may be supplied at the lower speeds, (2) additional air may be admitted above the Venturi tube for high speeds, and (3) the throat area may be varied. Additional fuel may be supplied by the use of an additional fuel jet, which has constant flow maintained through it by gravity regardless of the suction in the Venturi. Some carburetors are provided with an auxiliary air valve so arranged that the increased suction at high engine speeds will cause the valve to open and admit additional air to dilute the mixture. For low speeds, the mixture which gives good results is 12 to 13 parts of air to 1 part of gasoline by weight. At high engine speeds, 15 to 17 parts of air to 1 part of gasoline are usually required for maximum thermal efficiency.

280. The Automobile Engine.—The automobile engine, or motor as it is frequently called, is one of the most highly perfected types of internal-combustion engines. The engine used almost exclusively for automobiles is of the multi-cylinder type operating on the Otto four-stroke cycle. The peculiar demands made by the public have made necessary the building of engines that will withstand considerable abuse. Lubrication, carburetion, cooling, ignition, etc., must be simple and unfailing for long periods of time without any attention.

Figs. 257 and 258 show the essential parts of the six-cylinder engine used in the Packard automobile. The cylinders are cast en bloc (all the cylinders in a single casting) and bolted to an aluminum base (crankcase). The engine has a detachable cylinder head, the advantages of which are apparent. The poppet valves (an inlet and an exhaust valve for each cylinder) are opened by rocking levers, operated by cams, and are closed by springs. The camshaft is designed to open and close all of the valves at the proper time, so as to give the proper sequence of events in the six cylinders.

When the piston of one of the cylinders moves downward on the suction stroke the inlet valve is open, and the mixture of gasoline and air is drawn into the cylinder from the carburetor. The piston rises and compresses the mixture, both valves being closed. Compression serves

to mix more evenly the gasoline vapor with the air, the contour of the inside of the cylinder head, Fig. 258, also aids this mixing. As the pressure of the mixture increases during compression, the temperature also rises and good ignition is made easier. At the beginning of the next downward stroke the explosive mixture is ignited by an electric spark from a spark plug installed in the top of the cylinder. The combustion resulting from this ignition increases the pressure from about 60 to 300 lb. per sq. in. On the explosion or expansion stroke the piston is forced downward and by means of the connecting rod the driving force is trans-

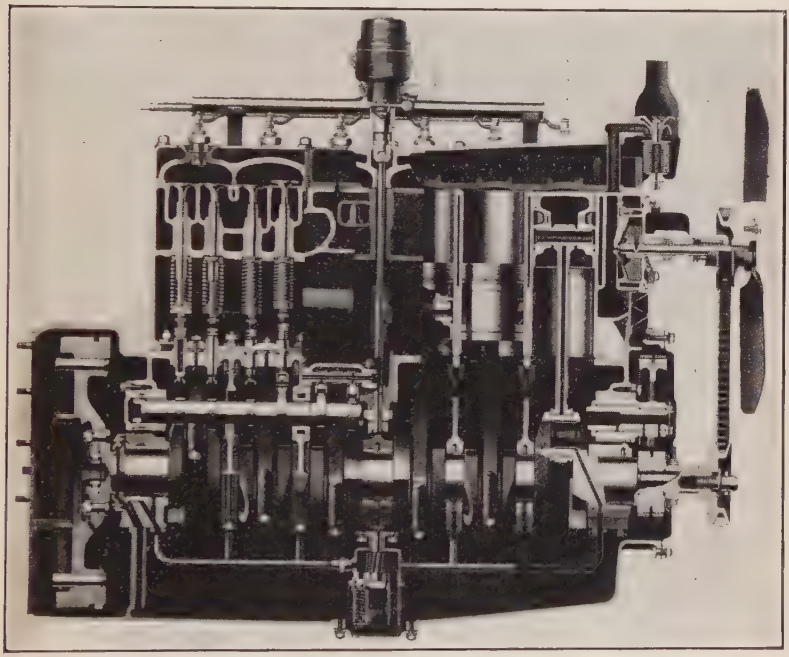


FIG. 257.—Side sectional view, Packard six-cylinder automobile engine.

mitted to the crankshaft of the engine. As the engine has six cylinders, the torque on the crankshaft is more uniform—there being three power impulses every revolution—than if only one cylinder were employed. On the next upward stroke of the piston the exhaust valve is open and the products of combustion are exhausted to the atmosphere; and the cycle of events, intake, compression, expansion and exhaust, is repeated.

In order to make the explosions in an engine cylinder as effective as possible, the ignition of the compressed mixture should occur a trifle before the piston reaches the top dead center. Ignition should take place, therefore, just before the end of the compression stroke. The

faster the engine is running the earlier in the stroke the ignition should occur, so as to insure complete combustion. That is, the "spark" should be further advanced (occur earlier) at high than at low speeds.

In an automobile engine the intake pipe between the carburetor and the engine is made with as many branches as there are cylinders. These branches of the intake pipe are called the intake manifold. In some recent engines the branches of the manifold going to the intake

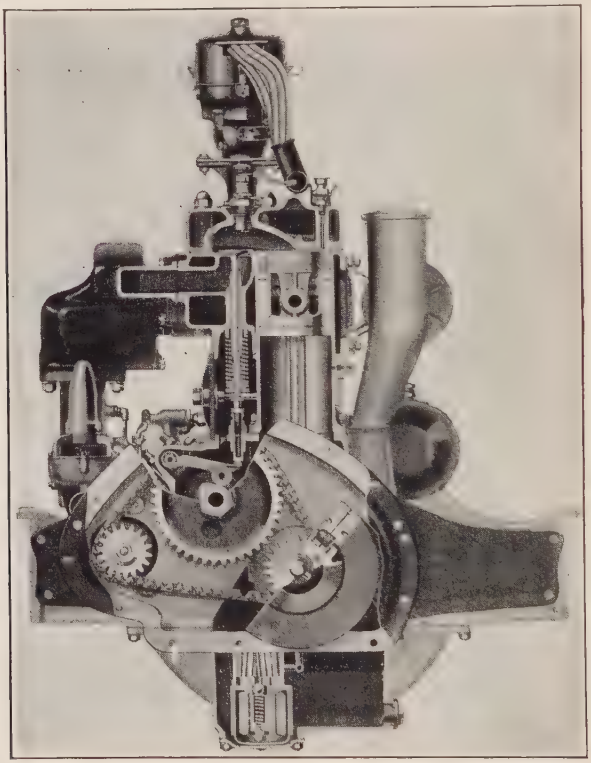


FIG. 258.—End sectional view, Packard six-cylinder automobile engine.

valves are cast in the sides of the cylinders or surrounded by the exhaust manifold. The advantage of this construction is that the manifold can be kept warm by the heat of the engine or exhaust, thus insuring better vaporization of the gasoline.

Cooling of the automobile engine is generally effected by a fan and water-jackets. Water for cylinder cooling is circulated through the water-jackets and radiator by means of a small rotary pump, located at the rear end of the fan shaft, Fig. 257. The cooling-water circulation

may be controlled by a thermostatic valve, Fig. 257. When the engine is cold, there is no circulation to the radiator (top connection). When the water in the jackets becomes heated the valve opens, and the circulation is through the radiator.

A gear pump, as illustrated in Figs. 257 and 258, is located at the lowest point of the crankcase and forces lubricating oil from the reservoir through a system of branch pipes to all parts of the engine and auxiliaries. The pump is operated by a shaft driven through a spiral gear on the camshaft and being submerged in the oil supply requires no priming. Before entering the pump, the oil is strained through a cylindrical screen which surrounds the entire pump. This strainer may be cleaned by removing a small plate on the bottom of the crankcase.

281. Oil Engines.—An oil engine may be defined as an internal-combustion engine that uses oil for fuel, the oil being sprayed into the cylinder or combustion chamber, during or at the end of the compression stroke. Oil engines are of three distinct types: the low-pressure, moderate-pressure and the Diesel (high-pressure).

The low-pressure type is so named because the compression is only 60 to 140 lb. per sq. in. Fuel is generally injected against hot plates, into a hot-bulb, or into an uncooled vaporizer chamber during the compression stroke. When the atomized fuel comes in contact with the hot surface, vaporization occurs and as the oil is usually kept out of the cylinder, only the expanding gases come in contact with the piston and cylinder walls. Initial vaporization is often produced by artificially heating the chamber, and after starting the engine the surfaces retain enough heat from explosion to explosion to be continuously effective as a vaporizer. These engines operate on the Otto cycle and the pressures resulting from the explosion are not high, seldom exceeding 250 lb. per sq. in.

Moderate-pressure oil engines employ compression pressures of 200 to 350 lb. per sq. in. These engines may be designed to operate on cycles very closely approximating the Otto or Diesel, or a combination of the two cycles. Ignition in this type of engine is usually secured by methods similar to those employed in low-pressure oil engines. Injection of the fuel cannot occur during the compression stroke as in the low-pressure type, on account of probable premature ignition. The temperature corresponding to 250 lb. per sq. in. pressure is sufficient to ignite most fuel oils, but this temperature is not high enough to insure perfect ignition at every stroke. In order to avoid occasional pre-ignition, the fuel oil in moderate-pressure oil engines is injected at the end of the compression stroke.

Fig. 259 illustrates a typical moderate-pressure oil engine, which

operates on the four-stroke Otto cycle having direct injection of the fuel oil. On the suction (outward) stroke of the piston the intake valve is open and air is drawn into the cylinder. Several degrees past the outer dead center the intake valve is closed and during the remainder of the inward stroke the piston compresses the air into the combustion chamber. Before the inner dead center is reached the fuel pump plunger is actuated by a cam, and two sprays of oil are injected into the combustion chamber. These sprays come together near the center of the combustion space and the force of their impact materially aids atomization and vaporization. Ignition takes place automatically due to the heat

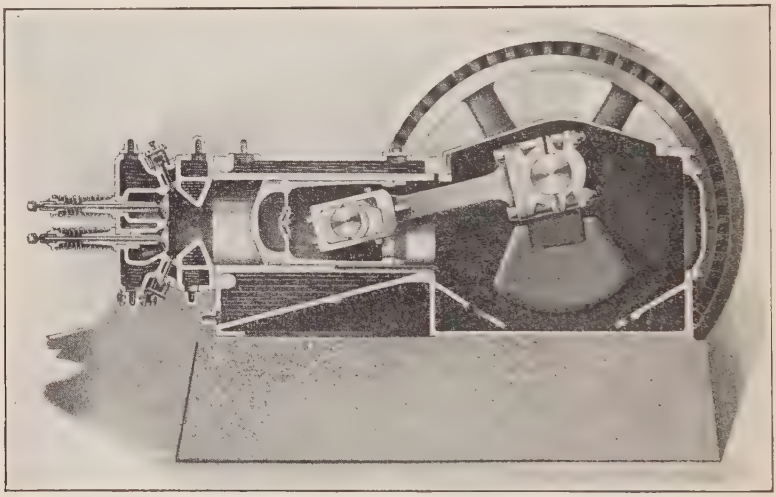


FIG. 259.—Ingersoll-Rand type "PO" oil engine.

of compression of the charge of air. The fuel injection is so timed that combustion at constant volume increases the cylinder pressure, which is maintained until about 20 deg. after admission. Expansion then begins forcing the piston toward the right; this is the working stroke from which the engine develops power. The piston passes outer dead center and during the return stroke forces the burned gases out through the exhaust-valve opening. The exhaust valve remains open until after the piston has just passed the inner dead center. The suction stroke of the next cycle begins and the events are repeated. The flywheels (considerably heavier than for steam engines) absorb sufficient energy during the working stroke to maintain a nearly uniform rotary speed of the crankshaft during the suction, compression and exhaust strokes.

282. The Diesel Engine.—The Diesel engine has been developed, both as a four-stroke and as a two-stroke cycle engine, exclusively as

an oil engine, and operates on a cycle, Fig. 253, which approximates the ideal Diesel cycle discussed in Art. 276. Of all types of commercial internal-combustion engines, Diesel engines generally give the best performance, and are used for stationary, marine, and locomotive purposes, especially in the larger sizes. The charge of air is compressed to a high pressure (about 500 lb. per sq. in.), and to a temperature of about 1000 deg. fahr. At the end of compression, the temperature is sufficiently high to ignite the fuel when injected into the cylinder in the form of a finely divided spray. Combustion of this fuel results at constant pressure; there is no explosion.

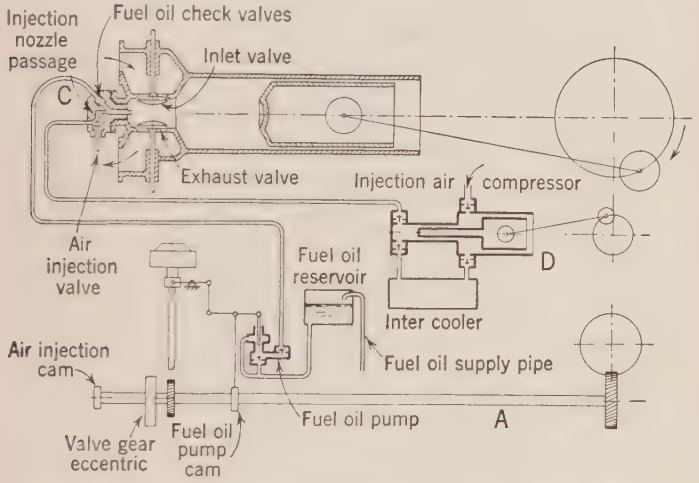


Fig. 260.—Diagrammatic arrangement of a Diesel engine plant.

When *air-injection* of the fuel is employed the pressure of the air for the blast must be considerably higher than the highest cylinder compression pressure. The fuel is injected with a blast of air at a high velocity, and is thus atomized and distributed through the main mass of compressed air in the cylinder. The injection pressure varies from 700 lb. per sq. in. for light loads to 1400 lb. for maximum loads. The auxiliary air is generally compressed in a two- or three-stage compressor driven directly by the engine; the compressor uses 5 to 15 per cent of the power developed.

The elements of a Diesel engine power plant using air-injection of the fuel are shown in Fig. 260. The engine illustrated is of the horizontal type operating on the four-stroke cycle. To the crankshaft is geared a layshaft *A* which revolves at one-half the engine speed. On this layshaft are mounted the cams used to actuate the admission, exhaust and

fuel-injection valves. The fuel pump is also driven by layshaft A, the amount of fuel charge being under control of the governor. The fuel is deposited in the fuel valve C, out of which it is forced by the air charge at the proper instant. The air blast is supplied by the compressor D which is driven by a crank on the end of the engine shaft. In this diagram the air line is not supplied with an air receiver, and the air is delivered directly from the compressor to the fuel valve. In addition to the valves shown, one or more cylinders of an engine are provided with "starting valves," which are used to supply compressed air to these cylinders for starting the engine.

The power used by the injection-air compressor may be eliminated, the mechanism of the engine simplified and the first cost decreased by using *solid-injection* of the fuel (also called airless injection or mechanical injection). Oil is forced into the cylinder at the proper time, through specially designed nozzles that produce the necessary fineness of spray, by one or more pumps under a pressure of 3000 lb. per sq. in. or more.

To reduce the size, weight and cost for a given power, Diesel engines are built to operate on the two-stroke cycle. A number of recent installations of this kind have shown fuel economies (lb. of oil used per b.hp. per hr.) approximately the same as with four-stroke engines, especially in the larger sizes. Fig. 261 shows a sectional view of a two-cycle solid-injection Diesel engine built in sizes from 60 to 540 brake hp. The "crank" end of the cylinder acts as a scavenging pump; this design requires the use of a piston rod and crosshead. The piston and piston rod are made hollow and oil under pressure is used for cooling the piston. The down stroke of the piston compresses the air on the under side of the piston and the air is delivered to receiver R, passing through "Feather" valves. An inclined port is employed to give the proper direction to the scavenging air as it blows into the cylinder at a pressure of about 3 lb. per sq. in. The exhaust ports E are uncovered by the piston near the end of the expansion stroke and exhaust begins. The piston next uncovers the inclined air port and when the pressure within the cylinder falls below that of the pressure in R, air enters to scavenge the cylinder and to provide a charge of air for the compression stroke. On the upward stroke of the piston the charge of air is compressed in the upper end of the cylinder to a pressure of nearly 500 lb. per sq. in. and air is drawn into the lower end of the cylinder.

Two-stage or divided combustion is used in this engine, Fig. 261. The fuel pump injects oil in an atomized form into the pre-combustion chamber, top of cylinder Fig. 261, just before the end of compression. The fuel is partially burned in this small space during injection, then, is

blown out into the combustion space of the power cylinder to be completely consumed in the main air charge at nearly constant pressure. The movement of the fuel from the first stage to the second stage is determined and controlled mainly by the downward movement of the

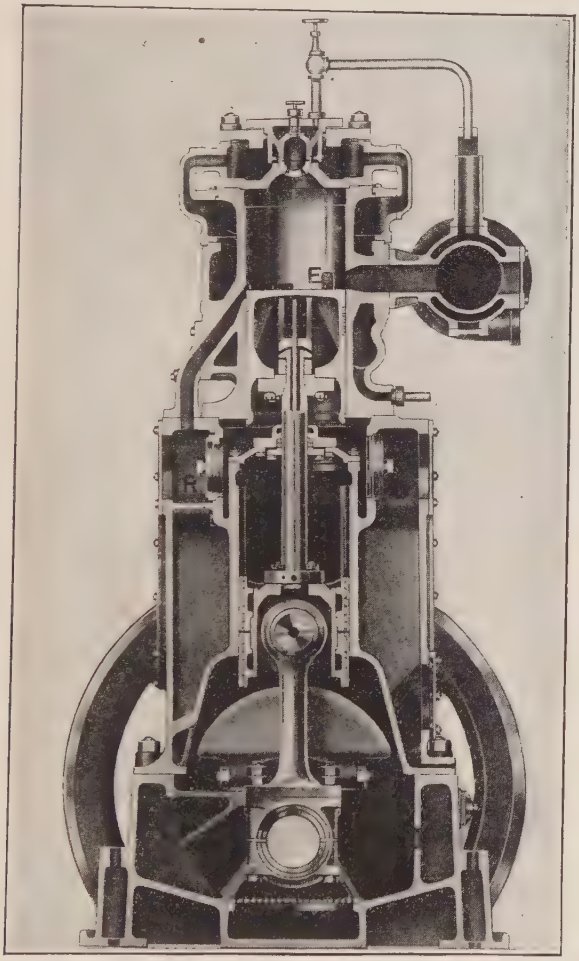


FIG. 261.—Section of a Worthington two-cycle solid-injection Diesel engine.

piston, which causes a differential pressure between the small spherical chamber and the top of the cylinder. However, the spraying of the fuel into the cylinder is started and assisted by the pre-combustion that takes place during injection. Combustion in this type of engine is fixed by the size and shape of the injection (pre-combustion) chamber, the outlet nozzle arrangement and the spray. Two-stage combustion has

the advantage of limiting the rate of burning of the oil during injection, so that accurate timing and careful injection control are not necessary. In the timing of direct mechanical injection, wherein the fuel meets the whole air charge at once, the combustion control required is represented by a few degrees of crank angle.

Diesel engines may be built double-acting in order to further reduce the size, weight and cost for a given power. The usual arrangement employs piston rods and crossheads, the pistons and rods being made hollow in order that oil or water under pressure may be used for cooling. The high temperatures to which the pistons in Diesel engines are subjected limits their diameter to about 20 in. unless some means of cooling is employed. Engines have been built to develop 2000 hp. per cylinder, with water-cooled pistons as large as 35 in. in diameter, in the double-acting two-stroke Diesel cycle types.

283. Governing.—The governing of internal-combustion engines does not differ materially from the governing of steam engines. The purpose of engine governors is to maintain approximately constant speed of rotation and to obtain the maximum thermal efficiency at all loads.

Automobile and marine engines are usually hand-governed. There are three general types of internal-combustion engine governors: (1) hit-and-miss governors, (2) quantity governors, and (3) quality governors.

With the *hit-and-miss* type of governor there is a working stroke for every cycle under conditions of maximum load. At lighter loads, when the speed increases, the governor mechanism acts to prevent admission of a charge of fuel occasionally and there is no explosion, causing the engine to " miss." When a " miss " occurs not only does the inlet valve remain closed, but the exhaust valve is generally held open, so that, during the strokes corresponding to the ordinary cycle the piston pumps exhaust gas into and out of the exhaust pipe with practically no loss of energy. The loss of a power stroke decreases the speed of the engine; the governor mechanism opens the inlet valve, an explosion or a " hit " occurs and the engine receives a power impulse. With this method of governing the engine operates either under conditions of maximum efficiency, or does not fire at all. Hit-and-miss governing gives better economy at light loads than the other methods, but does not give close speed regulation. Considerable intervals of time may intervene between power strokes of the engine and thus a heavy flywheel is required to avoid considerable variation in the speed. Hit-and-miss governing may be used for engines which do not require close speed regulation and with the smaller sizes, less than 40 brake horsepower.

Quantity governing may be accomplished by varying the amount of

mixture entering the cylinder, while the proportions of the mixture and the number of cycles are constant. This result may be obtained in one of two ways: (1) Cutting off the charge before the piston reaches the end of the suction stroke. (2) Throttling, except at full load, the charge during the suction stroke to reduce the amount of mixture entering the cylinder. The compression varies with the charge taken into the cylinder; reducing the compression lowers the efficiency, hence quantity governing is not as efficient for a large variation of load as the hit-and-miss type.

The *quality* governor varies the proportion of fuel to air, but the number of cycles and the quantity of charge per cycle remain the same. At full load a rich mixture is used, and at low loads a lean mixture. Mechanically, this method is quite simple but low thermal efficiencies result at light loads. Difficulty may be encountered in securing satisfactory ignition with very rich or very lean mixtures. Such mixtures burn very slowly and combustion continues in some cases throughout the entire expansion stroke.

A number of combinations of the foregoing governors have been used. Thus quality governing may be used for the high and medium loads, and the hit-and-miss governor for the light loads. Close regulation may be obtained by employing quality governing at high loads and changing to quantity governing at low loads. These combinations are desirable, but they tend to complicate the valve gear and require accurate adjustments.

The automobile engine is hand-governed by a quantity (throttle) control of the charge entering the cylinders, the proportion of gasoline to air remaining the same for a given carburetor adjustment, see page 390.

Oil engines use quality governing; the same amount of air being drawn into the cylinder during each suction stroke, with the governor controlling the amount of fuel injected per cycle. The amount of fuel forced into the cylinder may be governed by varying the length of the stroke of the fuel pump or by by-passing the fuel so that more or less is injected into the cylinder.

284. Starting and Lubrication.—A steam engine is started by turning the steam under pressure into the cylinder. An internal-combustion engine cannot be started in this way because there is no pressure to be turned into the cylinder, unless compressed air is used. Large engines are started with compressed air in the same way that steam is used in a steam engine. Small engines are set in motion by an electric motor, by compressed air, or by hand; the starting operation continuing until the explosions of the fuel begin in the cylinder.

Lubrication is a matter of great importance in the operation of

internal-combustion engines, because the rubbing surfaces are generally dry, the temperature and pressure are both high, and the speed is high. Oil is usually forced by pump pressure direct to the crankshaft bearings and, by means of drilled holes in the crank web to the crank pins, thence through oil pipes attached to the connecting rods or through hollow connecting rods, see Fig. 257, to the piston pins. The pistons and cylinders may be lubricated by oil thrown from the crank ends of the connecting rods.

285. Performance.—The horsepower rating of an internal-combustion engine required for driving a machine is greater than that of a steam engine or turbine for driving the same machine. The gas engine as rated has generally no overload capacity, whereas a steam engine or turbine operates successfully at a 25 per cent overload. Internal-combustion engines are commonly rated on the maximum brake horsepower output, while steam engines are usually rated on the indicated horsepower at which the best steam economy is obtained.

The indicated horsepower of an internal-combustion engine is determined in the same way as for the steam engine. The mean effective pressure is determined from indicator diagrams, which show the variation of the pressure in the engine cylinder at all points of the cycle. The remaining data required for the calculation of the indicated horsepower are the number of explosions per minute and the dimensions of the engine cylinder.

$$i.hp. = \frac{p_m l A N'}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (186)$$

where p_m = mean effective pressure, lb. per sq. in.
l = length of stroke, feet.
A = cross-sectional area of cylinder, sq. in.
N' = number of *explosions* per minute.

When an explosion occurs during every cycle: (1) for a single-acting four-stroke engine, the number of explosions per minute will equal one-half the revolutions per minute of the crankshaft; (2) for a single-acting two-stroke engine the number of explosions per minute will equal the revolutions per minute of the crankshaft; (3) for a double-acting two-cycle engine the number of explosions per minute will be twice the revolutions per minute. As in steam engines, the total indicated horsepower of a multi-cylinder internal-combustion engine is the sum of the horsepowers developed in the individual cylinders.

The manufacturer invariably works on a brake horsepower basis, on account of the difficulties encountered in accurately measuring the indicated horsepower. The accuracy of the indicator diagram may be

effected by the high speed, high temperature and high pressure of the engine. Special indicators, such as optical and diaphragm types, are generally used to obtain indicator diagrams, when the engine speed exceeds 500 r.p.m. and for greater accuracy at lower speeds. The brake horsepower of a gas engine is determined in exactly the same way as for the steam engine.

$$b.hp. = \frac{2\pi rwN}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (187)$$

where r = length of the brake arm, ft.
w = the net weight on the brake, lb.
N = the number of *revolutions*, per minute.

The *mechanical efficiency* of an engine is the ratio of the brake horsepower output to the indicated horsepower developed.

$$e_m = \frac{b.hp.}{i.hp.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (188)$$

The difference between the indicated and brake horsepower is known as the mechanical loss or friction horsepower and includes the negative loop (pumping loss) of the indicator diagram.

The *thermal efficiency* based on the brake horsepower shows what fraction, or percentage, of the heat supplied was actually delivered by the engine.

$$e_t = \frac{2545}{W_f \times H_f} \quad . \quad . \quad . \quad . \quad . \quad . \quad (189)$$

where 2545 = B.t.u. equivalent of 1 hp. per hr.
W_f = weight of fuel supplied, lb. per b.hp. per hr.
H_f = higher heating value per lb. of fuel supplied.

The heat supplied to an internal-combustion engine in the fuel may be accounted for in a "heat balance." This shows the heat converted into work (brake horsepower); and the heat lost: in overcoming friction, exhaust gases, water-jacket, lubricating oil, radiation, leakage, etc.

The fuel consumption, measured in cu. ft. of gas per b.hp. per hr. or lb. of oil per b.hp. per hr., varies considerably with the different types and sizes of internal-combustion engines. The fuel used by engines of a given type decreases with increase of cylinder diameter; the decrease may be considerable as the size progresses. The single-cylinder engine is invariably the most economical for a given size. However, practical difficulties limit the size to which cylinders may be built without com-

plicating the design, so that multi-cylinder engines are used for large sizes.

The fuel economy and the thermal efficiency for a given engine will vary with the load, Fig. 262; these remain nearly constant from three-quarter load to full load. Other types of internal-combustion engines give results somewhat similar to those illustrated in Fig. 262.

Modern multi-cylinder gas engines are guaranteed to produce a brake horsepower hour for each 9000 to 10,000 B.t.u. supplied in the gas to the engine, corresponding to a thermal efficiency of 28.3 to 25.5 per cent. The modern automobile engine has a fuel economy of 0.60 to 0.80 lb. gasoline per b.hp. per hr. when operating at or near the rated

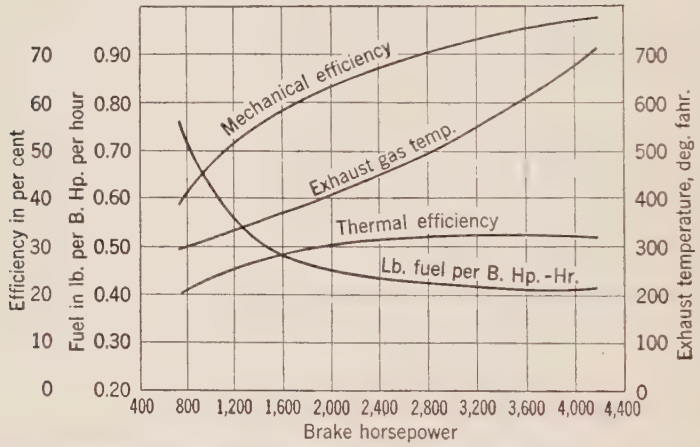


FIG. 262.—Performance curves of a 3750-B. hp. Diesel engine; 125 r.p.m., six-cylinder, single-acting, vertical, two-cycle, fuel-oil (18,740 B.t.u. per lb.), air-injection.

load and speed, with an air-fuel ratio of about 16 to 1 (by weight). Low-pressure (60- to 140-lb. compression) oil engines consume approximately 0.55 to 0.80 lb. of fuel oil per b.hp. per hr., whereas for moderate-pressure (200- to 300-lb. compression) oil engines, average values are 0.45 to 0.70 lb. of fuel oil per b.hp. per hr. when operated at from one-half to full load.

Some typical performance curves of a modern Diesel engine, operating under various loads, are shown in Fig. 262. From one-half to full load the fuel consumption varies from 0.46 to 0.42 lb. per b.hp. per hr. with a thermal efficiency of 29.4 to 32.2 per cent. Some Diesel engine tests have shown fuel economies of less than 0.40 lb. per b.hp. per hr. with a resultant thermal efficiency of 35 to 40 per cent. These figures give some idea as to the utility of internal-combustion engines and the future

possibilities for this type of prime mover, when considered from a standpoint of low fuel consumption and minimum operating cost.

PROBLEMS

1. A $9\frac{1}{2}$-in. $\times$ 19-in. gas engine is assumed to work on the ideal Otto cycle with adiabatic expansion and compression, receiving and rejecting heat at constant volume. The clearance volume is 272 cu. in. At the end of the suction stroke the pressure is 14 lb. per sq. in. abs., and the temperature of the charge is 210 deg. fahr. What is the air standard efficiency, and the temperature and pressure at the end of the compression stroke?

2. Calculate the ideal efficiency of a gas engine operating on the Otto cycle: (*a*) when the compression pressure is 120 lb. per sq. in. abs., and (*b*) when the compression pressure is 60 lb. per sq. in. abs. Assume the suction pressure to be 14.4 lb. per sq. in. abs. in each case and take $n = 1.35$.

3. Find the clearance volume in per cent of the stroke for an Otto cycle engine that will give an air-standard efficiency of 52 per cent. $n = k = 1.4$.

4. Determine the air-standard efficiency of an internal-combustion engine operating on the Otto cycle for clearance volumes of 5, 10, 15, 20, 30, and 40 per cent. $n = k = 1.4$. Plot a curve showing the variation of efficiency with clearance.

5. By a method similar to that used in finding the theoretical mean effective pressure of a steam-engine cylinder, find the theoretical m.e.p. of an engine operating on the Otto cycle, if the suction pressure is 14.5 lb. per sq. in. abs., pressure at the end of compression 87 lb. per sq. in. abs., and the pressure at the end of combustion 300 lb. per sq. in. abs. Use $n = 1.32$.

6. (*a*) Find the clearance volume required for a four-cycle Diesel engine if the pressure in the cylinder at the end of the suction stroke is 14.2 lb. per sq. in. abs., and at the end of compression 520 lb. per sq. in., use $n = 1.38$. (*b*) What will be the temperature at the end of compression, if the temperature at the beginning of compression is 150 deg. fahr.?

7. Find the air-standard efficiency of a Diesel engine having 6 per cent clearance when the fuel is injected for 2, 6, 10, 14, 18 and 22 per cent of the stroke. Tabulate similar to Table XIII using suitable headings; also plot a curve showing the variation of efficiency with fuel cut-off. Assume $k = 1.4$ and $n = 1.3$.

8. Calculate the air-standard efficiency for Diesel engines having clearance volumes of 3, 6, 9, 12 and 15 per cent, with fuel injection for 8 per cent of the stroke in each case. Use $k = 1.4$ and $n = 1.3$.

9. A 10-in. $\times$ 16-in. single-acting four-cycle gas engine runs at 200 r.p.m. The gross weight on the brake is 140 lb., the tare 30 lb. and length of the brake arm 60 in. The area of the indicator card is 1.20 sq. in. and the length 3 in. Scale of indicator spring, 200 lb. (*a*) Find the mechanical efficiency. (*b*) If 400 cu. ft. of city gas are supplied per hour, find the thermal efficiency based on b.hp. Higher heating value of the gas is 550 B.t.u. per cu. ft.

10. A $10\frac{3}{8}$-in. $\times$ 18-in. oil engine rated at 35 brake hp. uses 0.60 lb. of fuel oil per b.hp.-hr. at 250 r.p.m. The engine is four-cycle and single-acting. The fuel oil has a higher heating value of 18,500 B.t.u. per lb. Find: (*a*) the heat supplied in B.t.u. per b.hp. per min., (*b*) the m.e.p. based on rated b.hp., (*c*) the B.t.u. taken in per cu. ft. of suction displacement, and (*d*) the thermal efficiency.

11. A 12-in. × 21-in. gas engine has a rating of 50 b.hp. at 230 r.p.m. with a gas consumption of 12,000 B.t.u. per b.hp.-hr. The engine is four-cycle and single-acting. Find: (*a*) the m.e.p. based on rated b.hp., (*b*) the thermal efficiency, (*c*) the cost of operation per hour at full load, if the engine is supplied with gas, having a heating value of 520 B.t.u. per cu. ft., purchased at $0.80 per thousand cu. ft.; and (*d*) how this compares with the cost of operating a motor (to deliver 50 b.hp.) that is 75 per cent efficient, when power costs 4 cents per kw-hr.

12. The following data were obtained during a test of a 12-in. × 18-in. four-cycle single-acting oil engine:

Duration of test, 40 min.; oil used, 12.8 lb.; total revolutions, 8140; jacket-water, 740 lb.; rise of temperature of jacket-water, 75 deg. fahr.; m.e.p. in cylinder, 96 lb. per sq. in.; torque due to brake load, 784 ft.-lb.; higher heating value of fuel oil, 18,000 B.t.u. per lb. Find: (*a*) the indicated and brake horsepowers, and the mechanical efficiency; (*b*) the oil used per i.hp.-hr. and per b.hp.-hr.; (*c*) the heat converted into indicated work per minute; (*d*) the heat given to the water-jacket per minute; and (*e*) the heat lost by friction, exhaust gases, etc., per minute.

APPENDIX

ADDITIONAL TABLES

TABLE XIV

DENSITY OF WATER AT VARIOUS TEMPERATURES

Temp., Deg. Fahr.	Weight, Lb. per Cu. Ft.	Temp., Deg. Fahr.	Weight, Lb. per Cu. Ft.	Temp., Deg. Fahr.	Weight, Lb. per Cu. Ft.
20	62.37	80	62.22	180	60.58
30	62.42	90	62.11	200	60.12
40	62.43	100	62.00	250	58.83
50	62.42	120	61.71	300	57.32
60	62.37	140	61.38	350	55.46
70	62.30	160	61.00	400	53.42

TABLE XV

Areas of Circles

Diameter	Area	Diameter	Area	Diameter	Area	Diameter	Area
$\frac{1}{8}$	0.0123	10	78.54	30	706.86	65	3318.3
$\frac{1}{4}$	0.0491	$10\frac{1}{2}$	86.59	31	754.76	66	3421.2
$\frac{3}{8}$	0.1104	11	95.03	32	804.24	67	3525.7
$\frac{1}{2}$	0.1963	$11\frac{1}{2}$	103.86	33	855.30	68	3631.6
$\frac{5}{8}$	0.3068	12	113.09	34	907.92	69	3739.2
$\frac{3}{4}$	0.4418	$12\frac{1}{2}$	122.71	35	962.11	70	3848.4
$\frac{7}{8}$	0.6013	13	132.73	36	1017.8	71	3959.2
1	0.7854	$13\frac{1}{2}$	143.13	37	1075.2	72	4071.5
$1\frac{1}{8}$	0.9940	14	153.93	38	1134.1	73	4185.4
$1\frac{1}{4}$	1.227	$14\frac{1}{2}$	165.13	39	1194.5	74	4300.8
$1\frac{3}{8}$	1.484	15	176.71	40	1256.6	75	4417.8
$1\frac{1}{2}$	1.767	$15\frac{1}{2}$	188.69	41	1320.2	76	4536.4
$1\frac{5}{8}$	2.073	16	201.06	42	1385.4	77	4656.6
$1\frac{3}{4}$	2.405	$16\frac{1}{2}$	213.82	43	1452.2	78	4778.3
$1\frac{7}{8}$	2.761	17	226.98	44	1520.5	79	4901.6
2	3.141	$17\frac{1}{2}$	240.52	45	1590.4	80	5026.5
$2\frac{1}{4}$	3.976	18	254.46	46	1661.9	81	5153.0
$2\frac{1}{2}$	4.908	$18\frac{1}{2}$	268.80	47	1734.9	82	5281.0
$2\frac{3}{4}$	5.939	19	283.52	48	1809.5	83	5410.6
3	7.068	$19\frac{1}{2}$	298.64	49	1885.7	84	5541.7
$3\frac{1}{4}$	8.295	20	314.16	50	1963.5	85	5674.5
$3\frac{1}{2}$	9.621	$20\frac{1}{2}$	330.06	51	2042.8	86	5808.8
$3\frac{3}{4}$	11.044	21	346.36	52	2123.7	87	5944.6
4	12.566	$21\frac{1}{2}$	363.05	53	2206.1	88	6082.1
$4\frac{1}{2}$	15.904	22	380.13	54	2290.2	89	6221.1
5	19.635	$22\frac{1}{2}$	397.60	55	2375.8	90	6361.7
$5\frac{1}{2}$	23.758	23	415.47	56	2463.0	91	6503.9
6	28.274	$23\frac{1}{2}$	433.73	57	2551.7	92	6647.6
$6\frac{1}{2}$	33.183	24	452.39	58	2642.0	93	6792.9
7	38.484	$24\frac{1}{2}$	471.43	59	2733.9	94	6939.8
$7\frac{1}{2}$	44.178	25	490.87	60	2827.4	95	7088.2
8	50.265	26	530.93	61	2922.4	96	7238.2
$8\frac{1}{2}$	56.745	27	572.55	62	3019.0	97	7389.8
9	63.617	28	615.75	63	3117.2	98	7542.9
$9\frac{1}{2}$	70.882	29	660.52	64	3216.9	99	7697.7

TABLE XVI

Common Logarithms of Numbers

	0	1	2	3	4	5	6	7	8	9
100	0000	0004	0009	0013	0017	0022	0026	0030	0035	0039
101	0043	0048	0052	0056	0060	0065	0069	0073	0077	0082
102	0086	0090	0095	0099	0103	0107	0111	0116	0120	0124
103	0128	0133	0137	0141	0145	0149	0154	0158	0162	0166
104	0170	0175	0179	0183	0187	0191	0195	0199	0204	0208
105	0212	0216	0220	0224	0228	0233	0237	0241	0245	0249
106	0253	0257	0261	0265	0269	0273	0278	0282	0286	0290
107	0294	0298	0302	0306	0310	0314	0318	0322	0326	0330
108	0334	0338	0342	0346	0350	0354	0358	0362	0366	0370
109	0374	0378	0382	0386	0390	0394	0398	0402	0406	0410
110	0414	0418	0422	0426	0430	0434	0438	0441	0445	0449
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981

Note.—The Napierian logarithm of a number equals 2.3026 times common logarithm of that number or

$$\log_e n = 2.3026 \times \log_{10} n$$

TABLE XVI—*Continued*

COMMON LOGARITHMS OF NUMBERS

	0	1	2	3	4	5	6	7	8	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996

NOTE.—The Napierian logarithm of a number equals 2.3026 times common logarithm of that number or

$$\log_e n = 2.3026 \times \log_{10} n$$

TABLE XVII

Properties of Dry Air and Saturated Air *

Absolute Pressure, 29.92 Inches of Mercury

Temperature, Degrees F.	Weight of 1 Cubic Foot of Dry Air, Pound	Volume of 1 Pound of Dry Air, Cubic Feet	Volume of 1 Pound of Dry Air Plus Vapor to Saturate it, Cubic Feet	Weight of Water Necessary to Saturate 1 Cubic Foot of Dry Air, Pound	Weight of Water Necessary to Saturate 1 Pound of Dry Air, Pounds
0	0.08635	11.58	11.59	0.0000674	0.000781
10	0.08453	11.83	11.86	0.0001103	0.001309
20	0.08271	12.09	12.13	0.000177	0.002144
30	0.08104	12.34	12.41	0.000278	0.003444
32	0.08071	12.39	12.47	0.000303	0.003782
35	0.08019	12.47	12.55	0.000340	0.004268
40	0.07943	12.59	12.70	0.000410	0.005202
45	0.07861	12.72	12.85	0.000492	0.00632
50	0.07788	12.84	13.00	0.000588	0.00764
55	0.07710	12.97	13.16	0.000699	0.00920
60	0.07633	13.10	13.33	0.000829	0.01105
62	0.07604	13.15	13.40	0.000887	0.01188
65	0.07564	13.22	13.50	0.000979	0.01323
70	0.07491	13.35	13.69	0.001153	0.01578
72	0.07463	13.40	13.76	0.001229	0.01692
75	0.07424	13.48	13.88	0.001352	0.01877
80	0.07353	13.60	14.09	0.001580	0.02226
85	0.07283	13.73	14.31	0.001841	0.02634
90	0.07215	13.86	14.55	0.002137	0.03109
95	0.07153	13.98	14.80	0.002474	0.03662
100	0.07087	14.11	15.08	0.002855	0.04305
110	0.06964	14.36	15.73	0.003769	0.0593
120	0.06840	14.62	16.52	0.004920	0.0813
130	0.06720	14.88	17.53	0.006356	0.1114
140	0.06609	15.13	18.84	0.008130	0.1532
150	0.06498	15.39	20.60	0.01030	0.2122
160	0.06394	15.64	23.09	0.01294	0.2987
170	0.06289	15.90	26.84	0.01611	0.4324
180	0.06188	16.16	33.04	0.01991	0.6577
190	0.06094	16.41	45.00	0.02441	1.0985
200	0.05998	16.67	77.24	0.02972	2.2953
210	0.05931	16.86	.....	0.03594	.....
212	0.05910	16.92	.....	0.03732	.....

* G. A. Goodenough.

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